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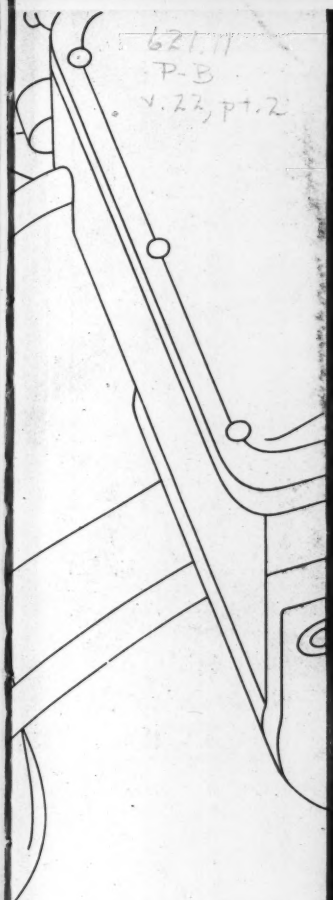
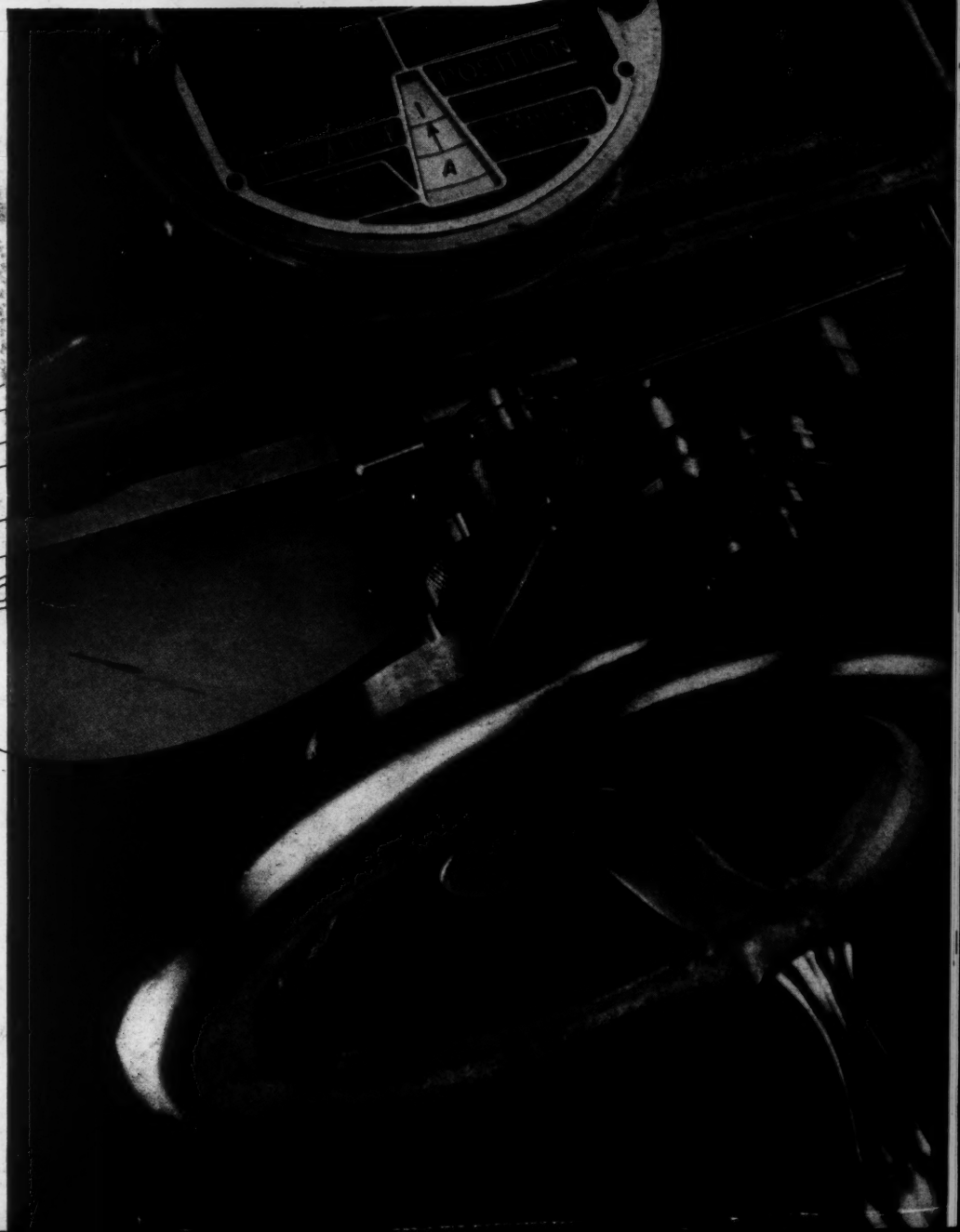
July 1950





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ENGINEERING ORGANIZATION
HIGH-SPEED CAMS
BARS IN TORSION
DESIGN ECONOMICS
GEAR STRENGTH AND LOADS

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WHAT ALLIS-CHALMERS 4-Point Motor Protection MEANS TO YOUR CUSTOMERS!

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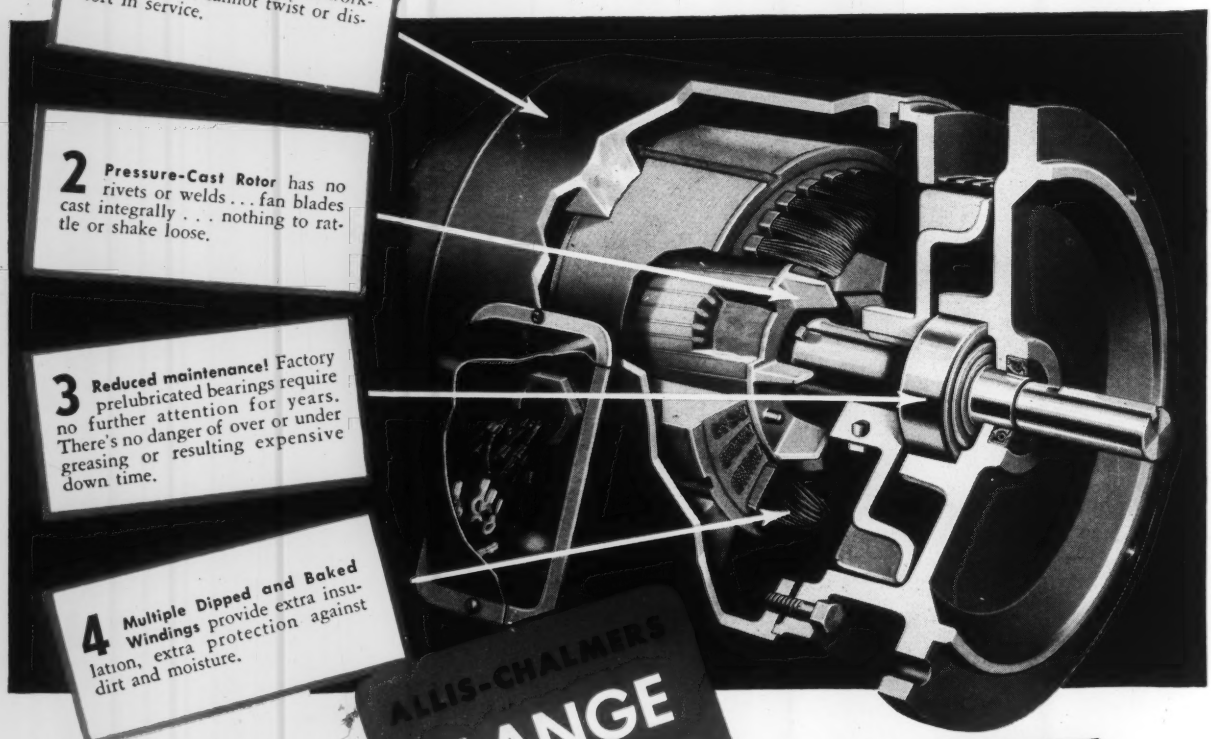

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- ☐ Flange Motor Specification Sheet 51S7234
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This Month's Cover: Positioning unit for accurately locating a machine table to successive positions which have been set up on a program record cylinder. Designed by Benzon Machine Co., the attachment utilizes precision switches to inform the operator when exact position is reached.

MACHINE DESIGN

Vol. 22—No. 7 July, 1950

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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Published by
THE PENTON PUBLISHING COMPANY
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G. O. HAYS .....*President and Treasurer*
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Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15. Copyright 1950 by The Penton Publishing Company. Acceptance under Act of June 5, 1934. Authorized July 20, 1934.

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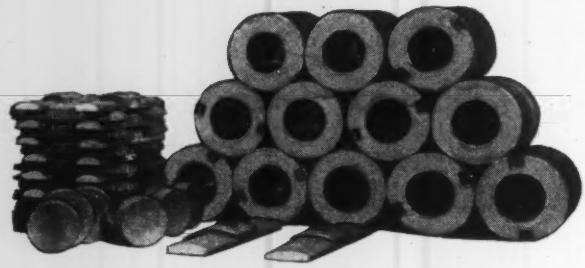
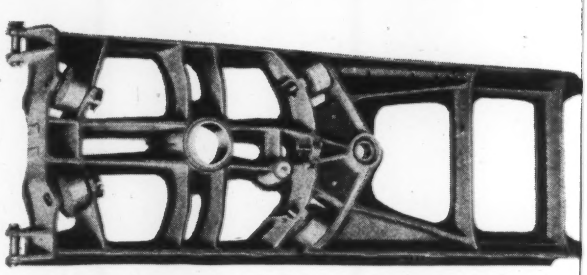
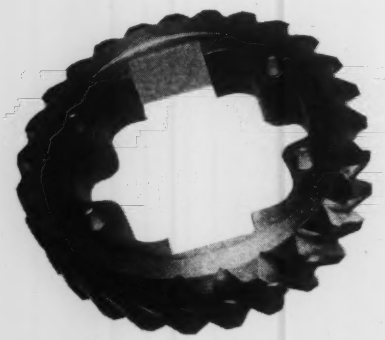
MACHINE COMPONENTS • ENGINEERING MANAGEMENT



Rolling Mill . . . Screw-down Nuts — Ampco alloys are selected for their high resistance to terrific pressures and impact. Rolling mill pressures are transferred through the screw to the nut threads. In cases such as a blooming mill, this pressure is applied with impact — increasing from zero to several million pounds in a fraction of a second. That's why they need the tough strength of centrifugally cast Ampco aluminum bronze.

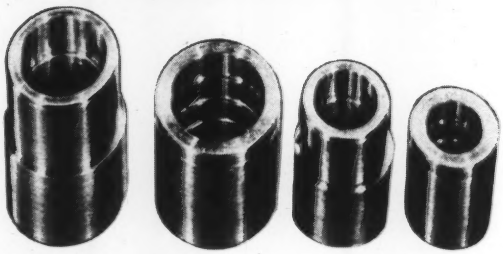


Earth-moving Equipment . . . Excavator Roller Bushings — Ampco Grade 16 selected for unusual wear resistance, excellent bearing qualities and high load-bearing strength. Turntable roller bushings must carry the tremendous weight of the cab, boom, and load. The turning is slow but eccentric because of the cantilever action of the boom and load. Ampco bronze alloys also used for gears, plates, cams, sleeves, and many other important excavator parts.



Garden Tractors . . . Worm Gears — Ampcoloy E-123 aluminum bronze selected for high strength and wear resistance, high impact and fatigue values. Reduced replacements to a mere 1/10 of the previous average. This outstanding saving is typical, accounts for the specification of Ampco aluminum bronzes for a wide variety of gears—from tiny fishing-reel gears to giant 1-ton gears for rolling mills.

Forging Machinery Upsetter Slides — Ampco Grades 18 and 20 selected by one manufacturer to replace hardened steel because they combine excellent bearing qualities with the necessary strength to withstand a 40,000 psi load. In two and a half years of service they showed little signs of wear. The same high physical properties make these alloys ideally suited for such applications as wear strips and wear plates.



Die Sets . . . Guide Pin Bushings — Ampco Grade 18 selected for its hardness and excellent bearing qualities. Seizing and galling are eliminated. Exceptional resistance to wear maintains tolerances. Correct alignment is assured regardless of speed of travel. Tests by one concern show Ampco guide pin bushings give 2½ times the life of previous material. Standard bushings are centrifugally-cast from Grade 18 Ampco Metal.

Machine Tools . . . Many Vital Parts — Over sixty leading tool manufacturers recognize the advantages of Ampco over ordinary bronzes. They specify Ampco because it assures long life through resistance to wear. Ampco Metal is also well known for its versatility, its hardness, its uniform quality, high impact strength, high yield strength and high compressive strength.

Ampco Extrusions . . . Rod, barstock, tubes . . . shapes — Produced in Ampco's own extrusion mill with a modern 2275-ton hydraulic press and complete processing equipment. Economical to use — saves metal and machining time and cost. Ampco extruded products have superior grain structure and exceptionally high strength values, *plus* close tolerances and good surface finish.

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4. Excellent bearing qualities
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6. Resistance to wear and/or corrosion
7. Little affected by extreme temperatures

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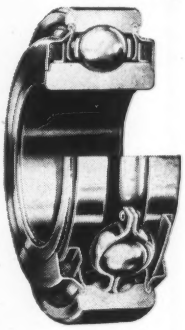

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BALL BEARINGS

NEW DEPARTURE • DIVISION OF GENERAL MOTORS • BRISTOL, CONNECTICUT

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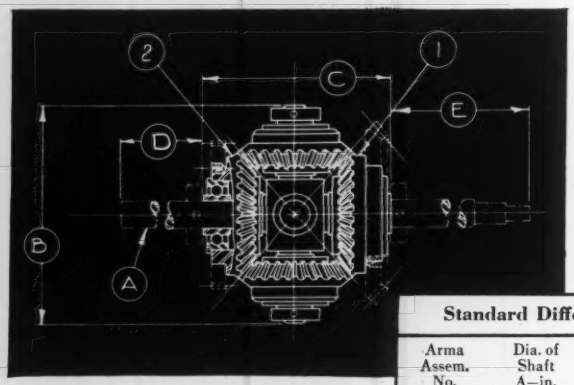
Now FOR INDUSTRIAL INSTRUMENTATION

ARMA MECHANICAL DIFFERENTIALS

To combine algebraically two mechanical quantities measured by angular displacement



Standard Differential furnished with overall length of approximately 6 in.

Arma Assem. No.	Dia. of Shaft A—in.	Dia. of Work. Cir. B—in.	Width C —in.	Details of Shaft Ends D & E	Pitch Dia. of Gears 1 & 2—in.	* BACKLASH—MINUTES OF ARC "F" Mesh	"C" Mesh	"P" Mesh
72044	3/16	2-3/16	1-1/8	TO CUSTOMER'S SPECIFICATIONS	1.187	24 ± 12	12 ± 6	6 ± 3
72045	1/4	2-1/16	1-23/32		1.062	26 ± 13	13 ± 6.5	6.5 ± 3
72046	5/16	2-7/16	1-7/8		1.312	22 ± 11	11 ± 5.5	5.5 ± 3
72047	3/8	2-13/16	2-11/32		1.500	18 ± 9	9 ± 4.5	4.5 ± 2
72051	1/2	3-1/16	2-5/8		1.687	17 ± 8.5	8.5 ± 4.3	4.3 ± 2
72052	5/8	3-1/2	2-7/8		2.000	14 ± 7	7 ± 3.5	3.5 ± 2

* Backlash (Min. of Arc.) measured on Gear #1 with Shaft "A" and Gear #2 locked.

DEFINITIONS OF ARMA DIFFERENTIAL MESHES

"F" Mesh = 0.002" ± 0.001 Backlash at Pitch Radius.

"C" Mesh = 0.001" ± 0.0005 Backlash at Pitch Radius.

"P" Mesh = 0.0005" ± 0.00025 Backlash at Pitch Radius.

New Differential folder just printed gives complete details.

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Construction Notes.

Dual ball bearings used throughout.

Shaft and spider are welded construction of stainless steel.

Spider gears and mating gears are of unlike materials—naval bronze and stainless steel.

Output gear hubs **permit wide range of end gear sizes.**

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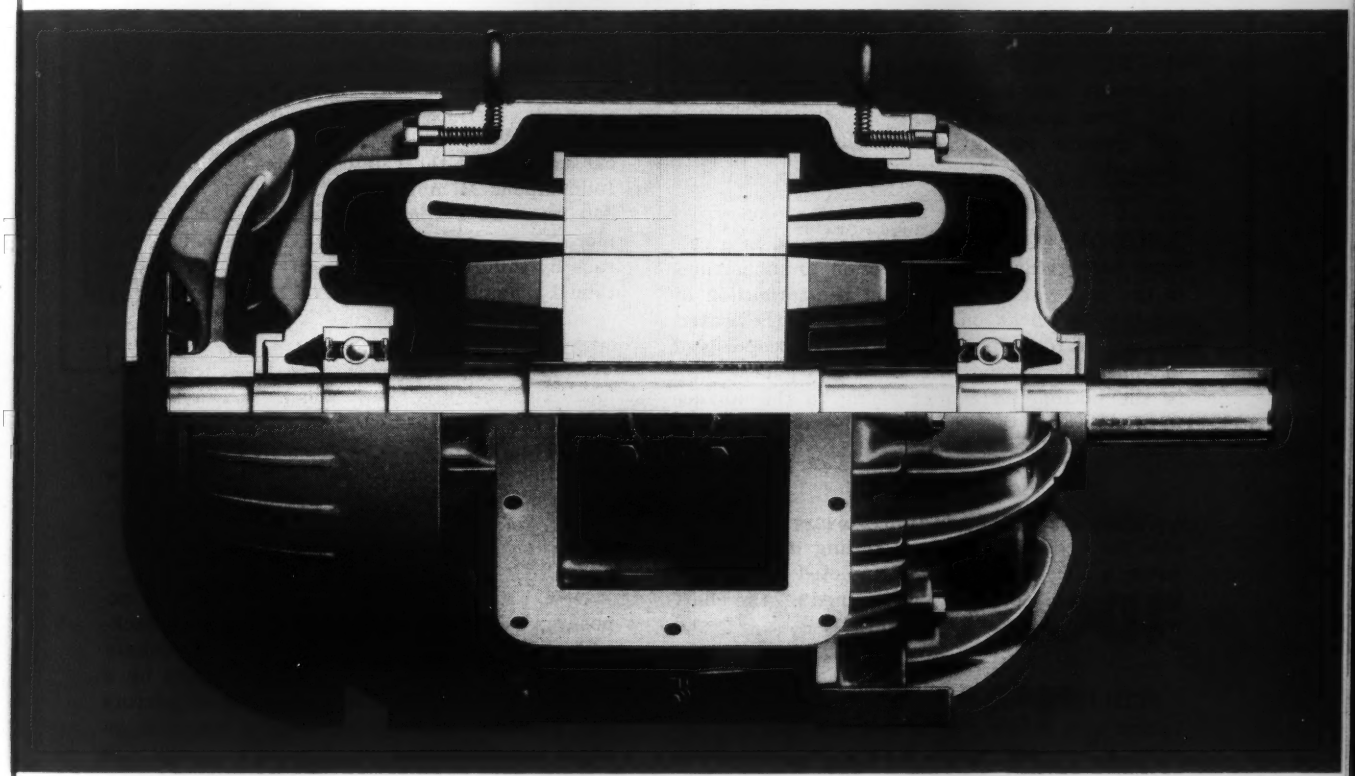

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CW-5

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Direct-Current Motors

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TOPICS

WELDBRAZING is a new joining method developed by the Uniworld Research Corp. In this process, less heat input than required for fusion welding is used with special steel filler alloys instead of the nonferrous alloys commonly used in brazing. Less destructive effect from localized heating than with welding and greater strength than with brazing are claimed.

POWDERED LEAD finely dispersed in a calcium base grease gave protection to submarines in the last war by preventing the formation of tell-tale oil slicks on the surface of the water. Developed for propeller shafts, periscopes and disappearing gun mounts, the heavy grease sank to the bottom instead of rising to the surface to give away the location of the submerged craft.

CIRCLE CHARTS of Vinylite are now available for repeated use in recording instruments. After a record has been completed and noted, the ink trace can be simply washed off the chart which may be reused indefinitely.

SCHLIEREN LIGHT SOURCE in a self-contained unit housing power supply, control circuits, lamps, and optical system has been developed by GE. The unit furnishes either continuous or high-intensity illumination for the photography of phenomena such as air-flow patterns around high-speed missiles and models of aircraft in wind tunnels. The well-known Schlieren system operates on the principle that a change in air pressure or temperature alters the light-bending properties of air.

PHOSPHORUS and cobalt or nickel alloys can be electrodeposited by a commercially feasible method developed recently by the Bureau of Standards. Plates of the alloys containing up to 15 per cent phosphorus are hard, corrosion resistant and bright.

INDUSTRIAL INJURIES, the National Safety Council reports, were substantially lower in both frequency and severity during 1949 as compared with 1948. The accident frequency rate was down 12 per cent, the severity rate, 9 per cent.

20-LENS HIGH-SPEED MOTION-PICTURE CAMERA for recording phenomena like underwater explosions and projectile behavior has been developed by the Navy. It will take up to 2000 frames per second on standard 35-mm film.

LIGHT-WEIGHT DIESEL ENGINE powered a car entered by the Cummins Engine Co. in the Indianapolis Memorial Day classic. Developing 340 hp at 4000 rpm, the engine weighs slightly more than 700 lb. Although forced out of the race by failure of an accessory part, the car attained a qualifying average of 129.208 mph.

MAGNESIUM is being used in a new fabric loom for the lay beam which carries shuttle blocks and shuttles in the weaving of materials such as elastic webbing. According to the Dow Chemical Co., looms equipped with this new light-weight beam require less power for higher-speed operation.

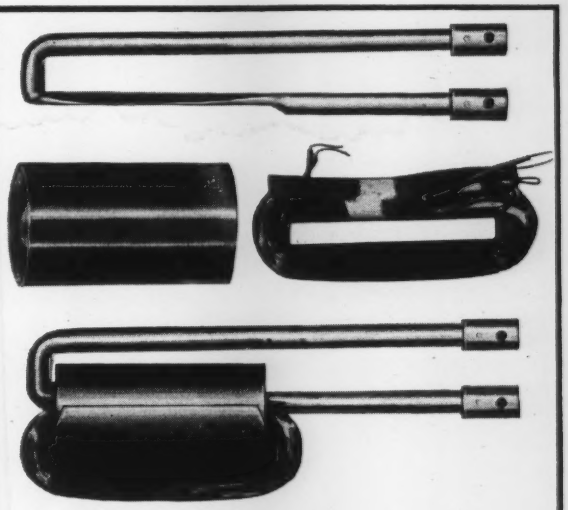
ONE 100-MILLIONTH SECOND time exposures have been used for both color and black-and-white photographs at the Army's Ordnance Proving Ground in Aberdeen, Md. Made by a camera that employs a new principle of mirrors and grids, photos will be used in research on shock and detonation.

HOT SPOT MACHINING removes metal three to 200 times faster than possible with machining at room temperatures. When the workpiece is heated just ahead of the cutting operation, additional advantages include production of a smooth continuous chip on most of the steels tested, absence of tool-wear difficulties and superiority of surface finish. Electrically induced heating is apparently more satisfactory than flame heating. Research proving the feasibility of the method is reported by the U. S. Department of Commerce.

AMERICAN STANDARDS approved during 1949 totaled almost double the average number approved annually during the five years of intense wartime activity. According to ASA, 268 standards were approved or revised during the year raising the total number of American Standards to 1121 and total revisions to 1014.

XUM

HEAVY, LESS EFFICIENT and slower to assemble, was the original Weller Soldering Gun composed of the parts shown. Compare these with the fewer parts at right in the new Gun, and you'll readily see why the new streamlined Weller is so popular.



THE NEW WELLER SOLDERING GUN handles 250 watts; heats, ready for use, in five seconds; has longer range to get into the tight spots, and is equipped with spot light. Uses current only when trigger is operated. It is assembled faster. Requires no bolts or nuts. The ⅜" Revere Copper Rod that replaces the secondary coil in the transformer is sheared, flattened, and bent at right angles in a 200-ton press in a single operation.

COPPER TREATS YOUR PRODUCT BETTER WHEN YOU

Control Your Temper

Revere Copper Rod replaces Secondary Coil in Soldering Gun Transformer . . . reduces number of parts, makes for a speedier, more efficient assembly . . . also makes possible a lighter, more compact unit of increased capacity.

WHEN the Weller Manufacturing Company, Easton, Pa., was completing the development of its new electric soldering gun, they were confronted with this problem: The ⅜" Revere Copper Rod used to replace the secondary coil in the transformer had to maintain its rigidity yet be sufficiently soft so that during the shearing, coining, and bending operations there would be no breaking or splitting of the rod.

Revere's Technical Advisory Service recommended a certain temper copper rod. It was discovered that Weller was getting a twist in the rod when it was installed in the assembled gun. Other tempers were tried and tested. Then a copper rod of a slightly harder temper than the first was recommended. That was it! Proper temper was the key. Proper temper was also the key to the .291 dia. copper rod used for the Soldering tip itself. For this, too, had to retain its rigidity and yet remain soft enough to be coined, punched, and formed without fracture.

"In addition to being extremely helpful in arriving at the proper tempers, Revere also recommended that we specify our rod in multiple lengths, and thus save considerably on scrap. They were also helpful in solving the problem of attaching the brass sleeve to the secondary rod in our Soldering Gun," the Weller Manufacturing Company tells us.

So you see, Revere's interest in your problem does not stop with the recommendation of its products. Perhaps Revere can help you. Why not take your current problem to the nearest Revere Sales Office and see?

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COPPER AND BRASS INCORPORATED

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Try the Guardian Series 335 D.C. Relay, one of hundreds of basic Guardian types. A general purpose relay with a wide range of applications, the Series 335 D.C. is built to the rigorous standard of aviation relays. It performs in accordance with the 10-G Vibration Test and the ANR-20-B test. Generous coil winding area permits single windings up to 15,000 ohms. Parallel and double windings are also available. Contact combinations are possible up to three pole, double throw.

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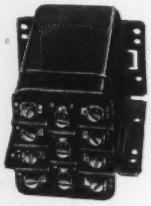

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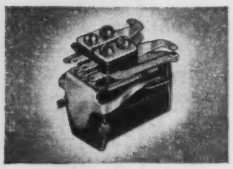

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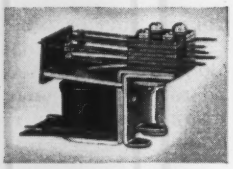

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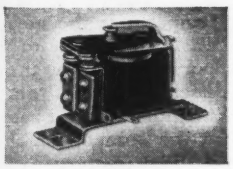

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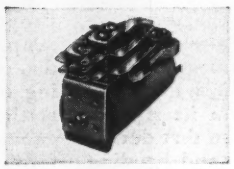
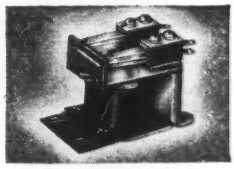


Series 695 D.C.



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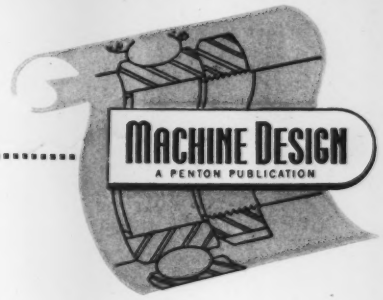


Education in Producibility

RETURN of highly competitive market conditions has emphasized the need for design that is planned for lowest-cost production. Achieving designs inherently economical to produce, however, poses a variety of problems.

In the past, the practice of many industries was to draw journeymen from the shop to fill the ranks in the engineering department. Thus an apprentice engineer, learning the fundamentals of machine design, brought with him a fair working knowledge of shop practices. This background led naturally to a degree of manufacturability in his designs. However, numerous developments have tended to eliminate this system and the vital problem of securing manufacturing economy must be solved today by more effective methods.

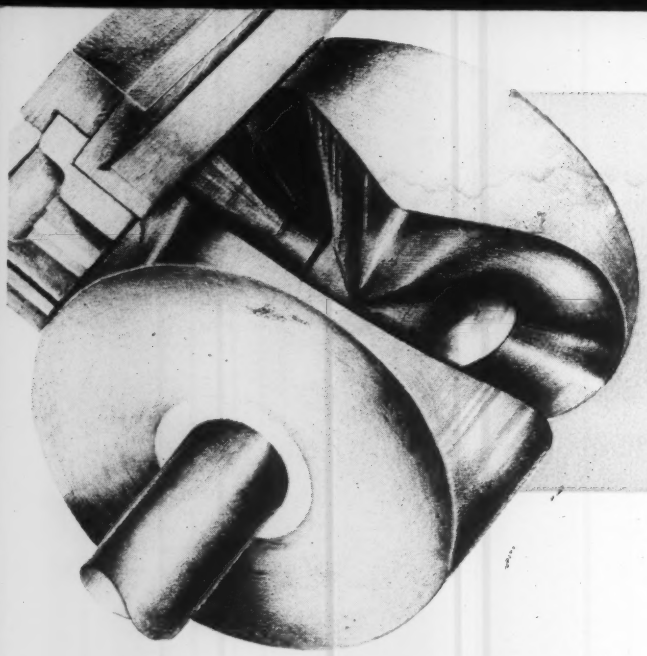
Although actual shop experience is of unquestioned value, a sound engineering background is the first requisite in design. Finding men with all-inclusive shop training, adequate for unbiased and thorough production design, is nearly impossible. Nevertheless, requirements that must be met to maintain leadership in competitive fields demand a real design-for-production understanding.

With prospect of substantial dollars-and-cents return, engineering management can cope with this problem by setting up continuous courses or planned sessions to school the designer's thinking and broaden his knowledge of design producibility. When the purpose of such a program is presented genuinely to designers, their wholehearted support and enthusiasm is assured. Design interpretation of manufacturing limitations can then be expected to gain advantages unrealized before.

Once it becomes clear that a design impossible of economical production is to be classed with one incapable of proper functioning, the real worth of production design study will assume its true significance. Engineering management has but to promote it properly.

H. A. Bolz

ASSOCIATE EDITOR



Design of Hi

. . . based on experimental tests a

By A. F. Gagne Jr.
Mechanical Development Engineer
Stratford, Conn.

MAXIMUM speeds of machines sometimes are determined by performance limitations of the cams used in their construction. Although designers have on occasion worked around such restrictions by substituting linkages, special gear trains, or other drives, modifications of the cam profile or minor construction changes may solve the problem. This article, without going deeply into theory, offers a few practical suggestions in this direction.

PROFILE: On certain cams, such as in screw machine feeds and computing mechanisms, the follower must be driven through a prescribed path which wholly determines the cam profile. For most applications, however, the cam is required only to move the follower smoothly from one position to another within a given time interval, and the designer can choose whatever profile he believes will best do the job.

Unfortunately there is a considerable disagreement among authorities as to which of the several common curves is ordinarily to be preferred. Handbooks recommend the constant acceleration-deceleration profile, also known as the gravity or parabolic cam, because the simplified analysis used shows this produces the lowest possible peaks of follower acceleration. Thus, is is assumed that the lowest possible inertia force loadings are encountered. A large number of successful machines have been built using cams of this type and, certainly, the designers might have made worse choices.

However, as has been brought out in recently pubished information*, the theoretical peak follower acceleration produced by a cam is only one of several important considerations. Very important is the rate of change of acceleration which can cause oscillatory deflections of the cam supports or oscillations between follower and driven mass, thereby producing loads several times higher than calculated. Also to be considered are cam cost, pressure angle, machine elasticity and damping, backlash, and the working loads: inertia, friction, gravity, spring, cutting tool pressure, etc. All these factors should be taken into account when the cam is limiting machine speed and performance or requiring excessive maintenance.

An example of the effect of excessive speed on a positive-motion parabolic cam is seen in *Fig.* 1. This nickel cast-iron barrel-cam segment was worn out after several years use in indexing a heavy turret. When the faces of this cam are studied, it is found that the acceleration half of the track is still in excellent condition, but the sides of the deceleration half are worn in waves over 1/64th-inch deep and resemble a washboard. The crests of one side face the valleys of the other. The reader can distinguish this pattern of wear by examining the edge of the unworn area below the path of the cam followers. It will be further noted that the badly worn areas are heavily pitted, comparable to the pitting visible in a fatigued ball bearing.

Backlash Caused Chattering

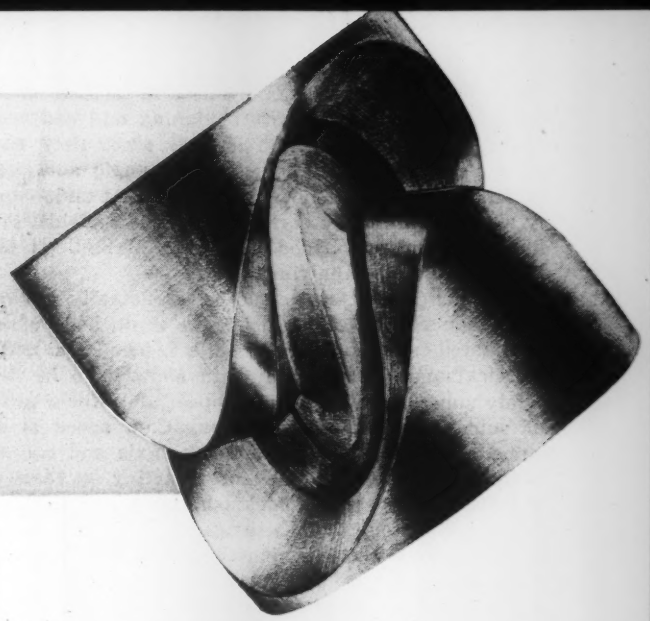
As might be imagined, this cam was noisy and shook the turret violently as it indexed. Even new cams with a perfect profile produced this chattering at the maximum operating speed, although to a lesser degree. The explanation lies in the abrupt inflection of the cam track made to reverse acceleration at midpoint. Because of backlash, the turret did not immediately start to slow down with the cam track, but continued at substantially constant velocity until all of the backlash was taken up. A violent impact resulted at this moment, and the resilience of the machine frame caused the turret to oscillate from one side of the track to the other. This oscillation, or chattering, continued some distance into the straightaway portion of the track, and naturally grew worse with time.

The high pressure angle of 40 degrees at midpoint permitted the shock forces to be transmitted back through the backlash of the resilience of the cam drive gearing and thus intensify the pounding. An effort was made to tighten up bearings and other clearances contributing to the backlash. This was only

*References are listed at end of article.

i-Speed Cams

abolic, harmonic, straight-line profiles



partly successful because certain clearances were essential to proper operation such as those between stud and cam roller, cam roller and track, and gear teeth.

Acceleration Change Should Not Be Abrupt

Study of other cam form possibilities was the next step. The objective was a cam with moderate peak acceleration and with no abrupt changes of acceleration. Still, because the parabolic cam had given no trouble with the sudden acceleration at start, high values or rapid changes of acceleration were considered permissible at zero or low pressure angles when there could be no reaction on the cam drive gearing. Another objective was low peak velocity, sought for reasons of machine and operator safety.

The simplest possible base motion curve is the straight-line. As seen in the displacement graph, *Fig.* 2, this profile offers the minimum peak pressure angle of any possible shape. Peak velocity is also low. Because of the theoretically infinite acceleration at the start and stop of motion, however, this profile cannot be considered except for the slowest speed machines. The acceleration is actually not infinite, of course, but rather has a finite limit determined by the elasticity of the mating parts and machine frame.

For practical purposes the violent start and stop can be tempered by small curvature as in *Fig.* 2. Like the straight-line profile, the modified straight-line has the special merit of a relatively low peak pressure angle, permitting the smallest cam practicable for a given pressure angle limitation. In view of the fact also that these cams can be generated readily on an ordinary milling machine, they are often used for moderate-speed applications, particularly when only one cam is needed.

In the case under consideration ease of machining was a minor consideration, but the modified straight-line appeared worth trying because of the low pressure angle and because it eliminated the sudden crossover of the standard parabolic. Since the customary radius approach gives a sharp peaking of follower acceleration, a parabolic approach to the straight-line was used instead, with the cam co-ordinates adjusted to ease the sudden acceleration changes of the parabolic form and provide maximum smoothness of action. The test was disappointing, for the cam was found to be actually somewhat rougher than a new parabolic cam. Apparently, the peak acceleration changes were occurring at sufficiently high pressure angles that they reacted back through the cam drive.

At a later date a trial was made of the harmonic curve, which had a somewhat higher pressure angle than the modified straight-line, yet offered promise because of its smooth change of acceleration at all values of pressure angle other than zero. Actually, the maximum pressure angle was substantially (17 per cent) less than the parabolic, and the calculated peak acceleration of the cam follower was only 23 per cent higher. Peak velocity was low, a desirable feature, and the acceleration drive torque (instantaneous product of the acceleration force reaction, the pitch radius, and the tangent of the pressure angle) was less than for any other profile considered.

The improvement was gratifying. The roughness and pounding were eliminated, and the index remained perfectly smooth and glide-like even at a speed 20 per cent over that used with the parabolic cam. A number of the harmonic cams have now been in use for several years at a 10 per cent speed increase without showing appreciable wear or increased noise.

Prior to this success, there had been some speculation as to whether the perfect cam for the application should not start with zero acceleration to take up backlash as gently as possible, build up quickly to a moderately high working value, ease off for a shockless cross-over as in the harmonic cam, and finally end abruptly, again as in the harmonic cam. The easy beginning is characteristic of two common curves, the cubic and the "t — sin t", better known as cycloidal. As seen in the table and in *Fig.* 2, the cubic is not advisable for ordinary applications because of its high values of peak pressure angle and peak acceleration. The cycloidal, on the other hand, appears to offer a useful combination of properties and indeed is highly recommended in recently pub-

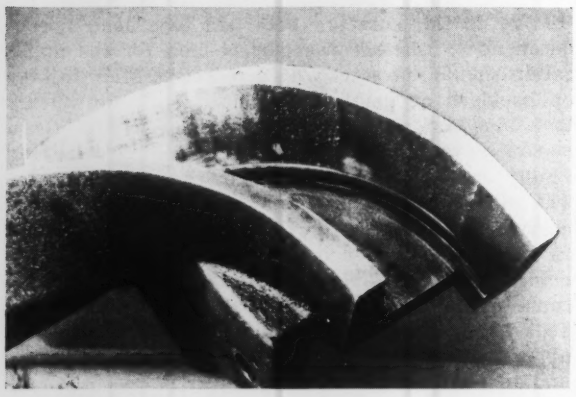
lished information. It was not used in the case under discussion because the soft beginning and end were not considered needed, particularly since they were accompanied by a higher pressure angle and peak velocity. It should be noted, however, that undoubtedly a trial would have been made of the cycloidal profile if recently-published experimental findings had been available at the time.

Whether or not the cycloidal is actually a cure-all remains to be seen since there is no body of accumulated experience on the subject. The latest mathematical and experimental data show it to be definitely superior but, it would appear, only for a group of somewhat limited conditions. Cam speed is assumed perfectly constant and cam axis and contour perfectly rigid despite the forces acting on them. It is assumed also that no backlash exists in the system. Given a machine in which the camshaft has negligible flywheel effect and in which the stiffness of the cam-follower coupling to the driven mass is relatively high and the stiffness of the cam support frame and drive are relatively low, a condition sometimes found in turret or dial cam indexes, it is possible that the cycloidal might prove inferior to some other profile. The cycloidal is definitely worth trying in troublesome applications, but more data are needed.

The following suggestions are made for selection of curves for positive cams.

1. For very low speeds and heavy loads, the modified straight-line is recommended for its ease of layout and manufacture and for its low pressure angle, which minimizes driving torque requirements for a given size and permits use of the smallest possible cam where pressure angle is the limiting factor
2. For average speed and load, the harmonic is recommended because it offers a practical compromise on cost, pressure angle, and rate of change of acceleration. Action may be improved if start and stop are slightly modified to give a ramping effect
3. The highest speeds suggest use of the cycloidal. This is based on the results of a limited set of experimental strain gage analyses, not on any actual machine usage known to the writer
4. Use of the parabolic is not recommended.

Fig. 1—Parabolic cam segment used to index a heavy turret. Wavelike wear pattern was caused by chatter. Deceleration half of cam faces front



Characteristics of Different Cam Motions
(Applied to index cam* shown in *Fig. 1*)

Type of Cam	Max Velocity of Follower (ft/sec)	Max Pressure Angle (degrees)	Max Accel. of Follower (ft/sec²)	Midpoint Accel. of Follower (ft/sec²)	Max Torque Required‡ (lb-in.)
Straight Line	0.453	22°50′	∞†	0	∞†
Modified Straight Line ..... (Parabolic Approach)	0.586	28°15′	3.60	0	12.2*M*
Modified Straight Line ..... (Radius Approach)§	0.628	30°	8.61	0	31.1*M*
Harmonic	0.708	33°22′	2.97	0	6.3*M*
Parabolic	0.902	39°58′	2.40	±2.40	12.6*M*
Cycloidal	0.902	39°58′	3.79	0	11.3*M*
Cubic	1.36	63°45′	7.21	±7.21	90.3*M*

* Index of 4.06 inches is required in ¾-seconds. PD of cam is 12½ inches. Time for acceleration = time for deceleration.
‡ Torque required of drive to overcome inertia of mass *M*.
† Theoretical: Actual values could not be determined without a strain gage or the equivalent.
§ Not shown in *Fig. 2*.

SPRING-LOADED CAMS: Spring-loaded cams introduce a new factor, the danger of the follower leaving the cam surface during the acceleration period of a fall. One method of overcoming this effect, aside from making the spring heavier, is to increase the acceleration period at the expense of the deceleration at the bottom of the fall. The corresponding adjustment is often made to control the deceleration period of a rise.

Inertia Forces Are Balanced

Another adjustment, for critical applications, is that of balancing the inertia forces against the change in spring load with displacement. The resulting cam curve is compounded of a parabolic function to balance the spring preload plus a cubic to balance the rise in spring load and is known as a "proportional" curve. In addition, arbitrary modifications should be made in the co-ordinates to give an easy cross-over and finish.

In the final analysis, the fundamental requirement of any cam, spring-loaded or positive, is a continuous motion curve suitably matching the applied loads, clearances, and inertias. There should be no abrupt changes in displacement or its successive derivatives of velocity, acceleration, and change of acceleration. By this token any empirical curve would be acceptable, and indeed the only serious objection thereto appears to be the difficulty of determining or reproducing such a function.

In certain well-established mass-production industries, such as in the manufacture of valve cams for high-performance internal combustion engines, the limitations of the standard cam formulas have justified extensive development of empirical motion curves. Here the designer uses a slow initial rise or ramp to absorb backlash, followed by a rapid acceleration limited principally by the safe column strength of the valve stem. A "proportional" acceleration is often used to maintain the sum of inertia and spring

forces at a constant value equal to the stem column strength. Deceleration and the return downward motion occupy comparatively long periods in order to prevent jump of the follower off the cam and to give easy seating, with the cam shape further modified to give adequate gas passage capacity. High-precision manufacturing methods are essential, however, to secure the full benefits of the development work, particularly in view of the small cam sizes.

Slight Profile Error May Be Serious

TROUBLE SHOOTING: On occasion the designer finds that the carefully planned profile does not live up to expectations; the machine is dangerously rough at the speed sought. He is advised to double check whether the cam profile is actually that intended, for slight errors can produce serious effects on the resultant accelerations. The cam co-ordinate chart is best checked by the well-known method of subtraction, in which the increments of displacement (velocity equivalents) for each degree are successively subtracted one from the next to give the acceleration equivalents. These differences should be constant for a parabolic curve and progressive for other types; any sudden jump marks an error.

The usual machine-shop methods of preparing cam masters, as layout, saw or mill, and file, are also prone to error. Peculiar humps and dips are not unknown in masters made by the finest shops. It is desirable therefore to make the master as large as possible, thus minimizing the effect of an error of given size, and to check the layout and finished cam with a drafting-room template.

In concluding this discussion of cams, it must be emphasized that correct profile alone will not insure a successful machine. The maximum pressure angle should be kept below 30 degrees, if necessary enlarging the cam. This has the additional merit of increasing the probable accuracy of machining. This accuracy will last longer if a relatively large roller follower is used to minimize local pressures and wear and if hardened inserts are used at critical points.

Preload and Cushioning May Solve Problem

Successful high-speed performance also demands that backlash be held to an absolute minimum, particularly as regards positive cams. The writer knows of several instances wherein a difficult problem was solved or a poor cam design salvaged by the substitution of preloaded, oversize antifriction bearings for inadequate bushed bearings. In one case preload plus cushioning supplied by rubber rings between ball thrust bearings was sufficient to enable a badly worn barrel cam to continue in production for a considerable period without further difficulty. Supplemental spring loading on the driven member helps take up backlash and can sometimes be used to buck working or friction loads and relieve the cam of part of its duty. The point of cross-over, which takes place when the resultant of all loads changes sign, should then be checked carefully because the tendency toward pounding will be accentuated rather than diminished if cross-over should occur at a point on the cam curve of rapid velocity change.

In case of severe oscillation, attention should be given to raising the natural frequency of vibration of the machine frame and drive and of the follower system, since resonance may be a factor. This is done by making all parts more rigid and by paring weight from the moving parts if this can be done without sacrificing rigidity or necessary strength.

A problem that does not yield to the methods discussed above can be attacked effectively by a return to first principles, an examination of the basic inertia force equation,

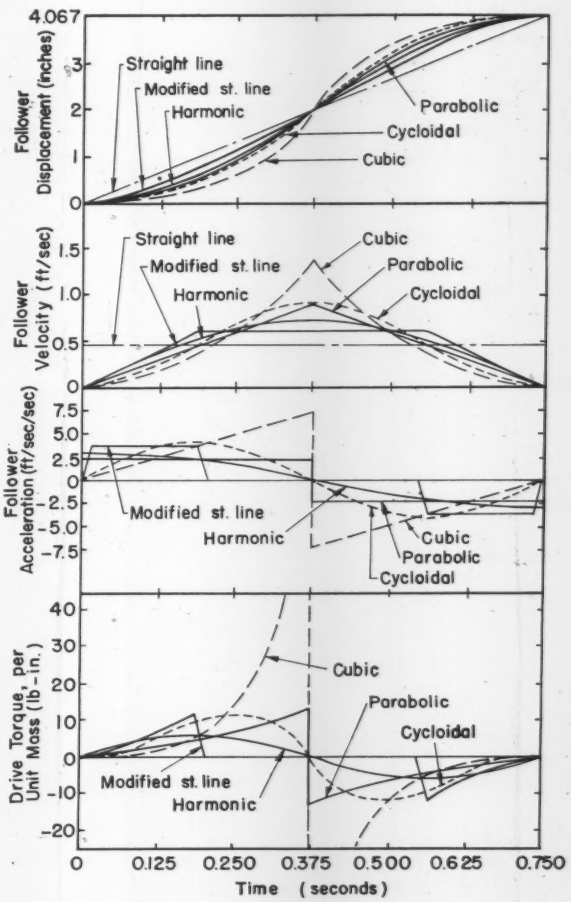
$$F = K\frac{Md}{t^2}$$

where M = mass, d = displacement and t = time.

As seen herein, proportionate benefits are obtained by reducing the weight of the accelerated parts, either by trimming off excess metal or by substituting stronger or lighter materials of construction. It is further helpful to reduce the working stroke to a practical minimum. The third element, time, the period of action, appears as a squared function and

(*Concluded on Page* 194)

Fig. 2—Comparison of common cam curves, including straight line, modified straight line, harmonic, parabolic, cycloidal, and cubic profiles



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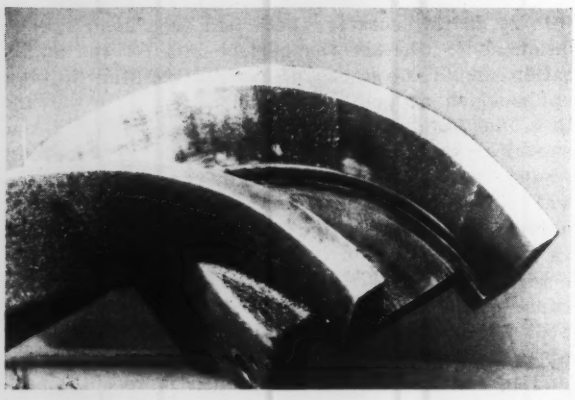

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Cubic	1.36	63°45′	7.21	±7.21	90.3*M*

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† Theoretical: Actual values could not be determined without a strain gage or the equivalent.
§ Not shown in *Fig.* 2.

SPRING-LOADED CAMS: Spring-loaded cams introduce a new factor, the danger of the follower leaving the cam surface during the acceleration period of a fall. One method of overcoming this effect, aside from making the spring heavier, is to increase the acceleration period at the expense of the deceleration at the bottom of the fall. The corresponding adjustment is often made to control the deceleration period of a rise.

Inertia Forces Are Balanced

Another adjustment, for critical applications, is that of balancing the inertia forces against the change in spring load with displacement. The resulting cam curve is compounded of a parabolic function to balance the spring preload plus a cubic to balance the rise in spring load and is known as a "proportional" curve. In addition, arbitrary modifications should be made in the co-ordinates to give an easy cross-over and finish.

In the final analysis, the fundamental requirement of any cam, spring-loaded or positive, is a continuous motion curve suitably matching the applied loads, clearances, and inertias. There should be no abrupt changes in displacement or its successive derivatives of velocity, acceleration, and change of acceleration. By this token any empirical curve would be acceptable, and indeed the only serious objection thereto appears to be the difficulty of determining or reproducing such a function.

In certain well-established mass-production industries, such as in the manufacture of valve cams for high-performance internal combustion engines, the limitations of the standard cam formulas have justified extensive development of empirical motion curves. Here the designer uses a slow initial rise or ramp to absorb backlash, followed by a rapid acceleration limited principally by the safe column strength of the valve stem. A "proportional" acceleration is often used to maintain the sum of inertia and spring

forces at a constant value equal to the stem column strength. Deceleration and the return downward motion occupy comparatively long periods in order to prevent jump of the follower off the cam and to give easy seating, with the cam shape further modified to give adequate gas passage capacity. High-precision manufacturing methods are essential, however, to secure the full benefits of the development work, particularly in view of the small cam sizes.

Slight Profile Error May Be Serious

TROUBLE SHOOTING: On occasion the designer finds that the carefully planned profile does not live up to expectations; the machine is dangerously rough at the speed sought. He is advised to double check whether the cam profile is actually that intended, for slight errors can produce serious effects on the resultant accelerations. The cam co-ordinate chart is best checked by the well-known method of subtraction, in which the increments of displacement (velocity equivalents) for each degree are successively subtracted one from the next to give the acceleration equivalents. These differences should be constant for a parabolic curve and progressive for other types; any sudden jump marks an error.

The usual machine-shop methods of preparing cam masters, as layout, saw or mill, and file, are also prone to error. Peculiar humps and dips are not unknown in masters made by the finest shops. It is desirable therefore to make the master as large as possible, thus minimizing the effect of an error of given size, and to check the layout and finished cam with a drafting-room template.

In concluding this discussion of cams, it must be emphasized that correct profile alone will not insure a successful machine. The maximum pressure angle should be kept below 30 degrees, if necessary enlarging the cam. This has the additional merit of increasing the probable accuracy of machining. This accuracy will last longer if a relatively large roller follower is used to minimize local pressures and wear and if hardened inserts are used at critical points.

Preload and Cushioning May Solve Problem

Successful high-speed performance also demands that backlash be held to an absolute minimum, particularly as regards positive cams. The writer knows of several instances wherein a difficult problem was solved or a poor cam design salvaged by the substitution of preloaded, oversize antifriction bearings for inadequate bushed bearings. In one case preload plus cushioning supplied by rubber rings between ball thrust bearings was sufficient to enable a badly worn barrel cam to continue in production for a considerable period without further difficulty. Supplemental spring loading on the driven member helps take up backlash and can sometimes be used to buck working or friction loads and relieve the cam of part of its duty. The point of cross-over, which takes place when the resultant of all loads changes sign, should then be checked carefully because the tendency toward pounding will be accentuated rather than diminished if cross-over should occur at a point on the cam curve of rapid velocity change.

In case of severe oscillation, attention should be given to raising the natural frequency of vibration of the machine frame and drive and of the follower system, since resonance may be a factor. This is done by making all parts more rigid and by paring weight from the moving parts if this can be done without sacrificing rigidity or necessary strength.

A problem that does not yield to the methods discussed above can be attacked effectively by a return to first principles, an examination of the basic inertia force equation,

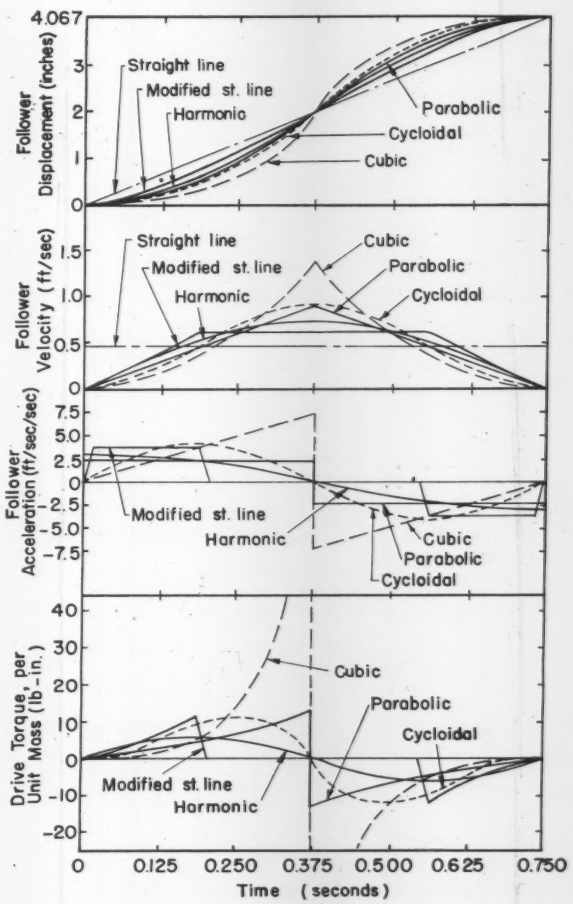
$$F = K\frac{Md}{t^2}$$

where M = mass, d = displacement and t = time.

As seen herein, proportionate benefits are obtained by reducing the weight of the accelerated parts, either by trimming off excess metal or by substituting stronger or lighter materials of construction. It is further helpful to reduce the working stroke to a practical minimum. The third element, time, the period of action, appears as a squared function and

(*Concluded on Page* 194)

Fig. 2—Comparison of common cam curves, including straight line, modified straight line, harmonic, parabolic, cycloidal, and cubic profiles

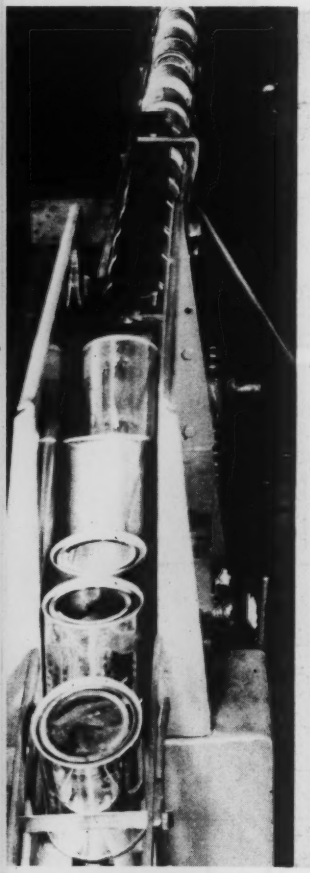


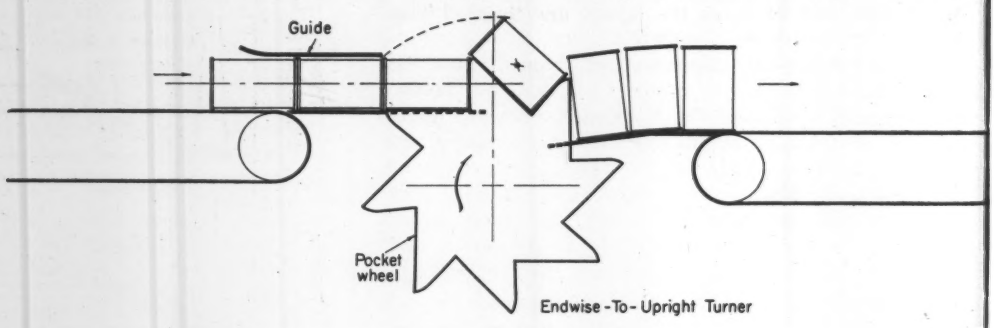
SCANNING the Field For Ideas

Handling of containers, such as cans during manufacture, often requires that they be turned in position on transfer conveyors between machines as shown below. To turn the cans either endwise-to-upright or upright-to-endwise, below, two simple vertical-starwheel units have been designed by the Chain Belt Co. The former unit operates without an outside power source. The cans enter a free turning, balanced pocket wheel and the pressure of the following cans plus the weight of the cans in the pockets turn the wheel through a 90-degree arc, depositing the cans in an upright position.

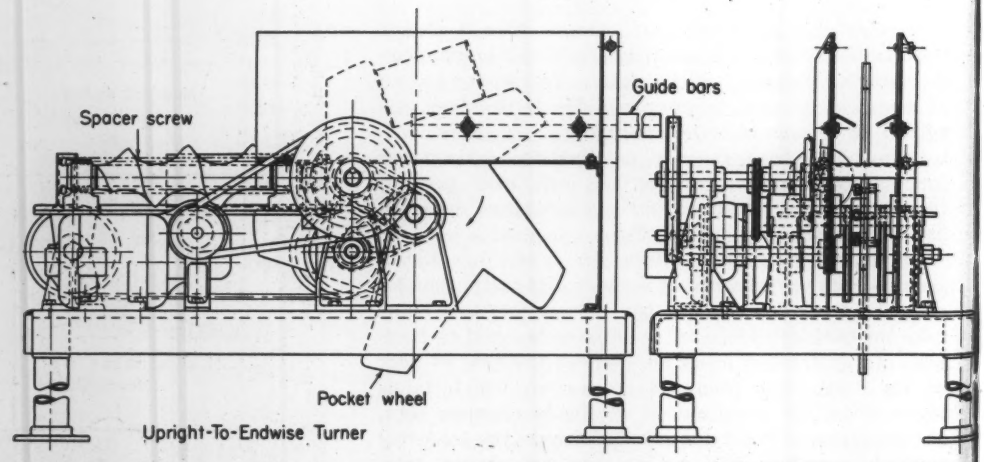
As shown in the simplified sketch, below, a top guide is provided at the entrance to the turner. This guides the cans into a straight line and "unlocks" them should one happen to rest inside the end flange of another. Also it prevents end flanges from locking into each other when a can is lifted by the wheel. When a can enters a pocket, the following cans push the wheel until the weight of the first can overbalances the wheel. This continues with the wheel speed governed by the speed of the cans entering it.

When cans are to be turned upright-to-endwise, the can turner shown at bottom is employed. This unit is powered by a 1/3-horsepower gearmotor, operating



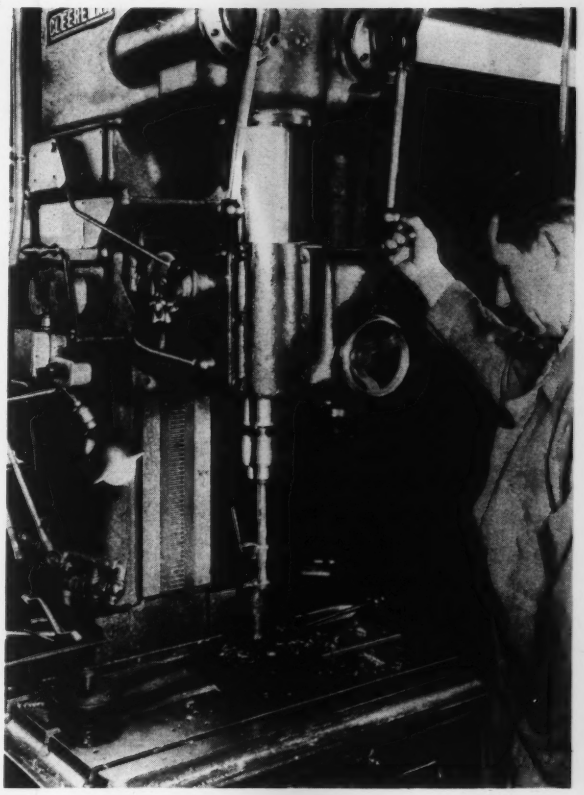


Endwise-To-Upright Turner



Upright-To-Endwise Turner

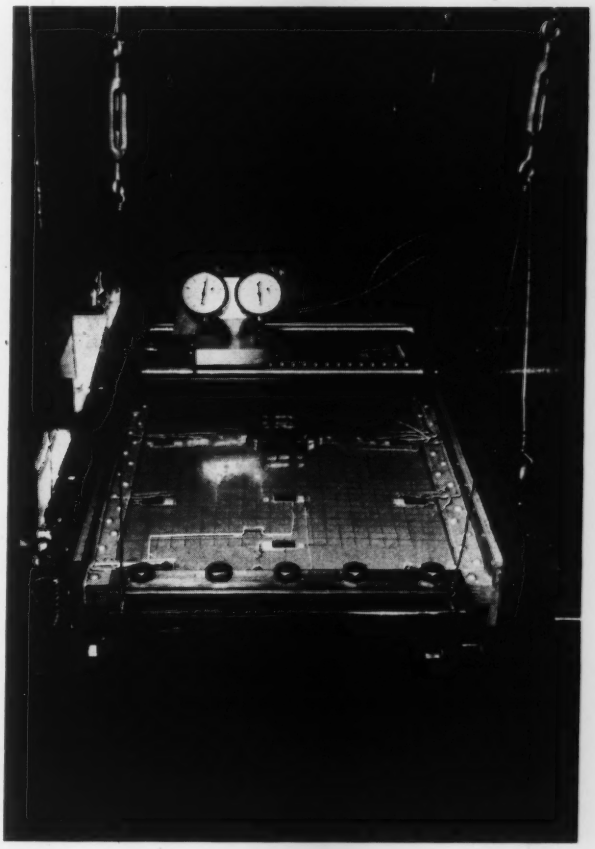
a spacer screw, pocket wheel and chain conveyor drive. Cans are fed into the turner in an upright position from the conventional conveying medium. As the cans enter the turner, they are conveyed forward on two strands of roller chain while being spaced by the spacer screw and are delivered in timed relation into the pockets of the rotating star-wheel. The wheel then lifts the cans from the conveyor chains and rotates them through a 90-degree arc and deposits them endwise on horizontal guide bars. As the wheel continues to rotate, the cans are pushed forward, out of the turner, before the next can is deposited on the guide bars.



Universal work spotter, right, for drill presses reduces setup time and gives a standard drill press the versatility of a radial drill. Arms of the attachment employ the pantagraph principle, allowing the operator to move the work freely on the work table. Developed by the Torkarm Corp., the attachment converts the torque of a spinning drill, tap or reamer into useful force which aids in holding work in place.

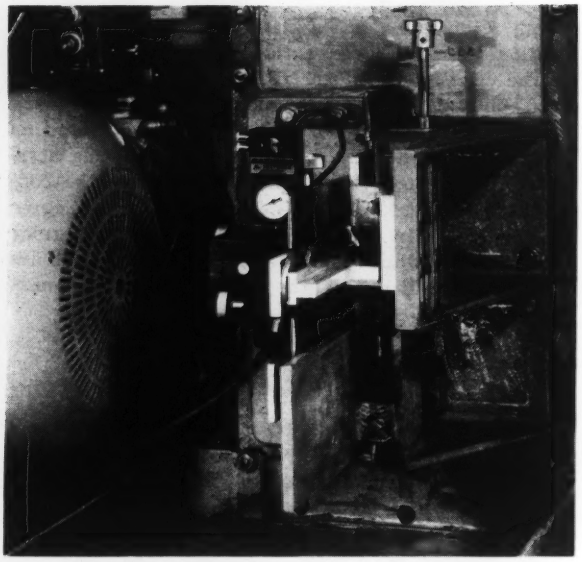
Reinforced openings in stressed skins of the wings and fuselages of aircraft often utilize metal rings riveted to the under surface of the skin. Because it has not been known how these reinforcements actually distribute stresses, the National Bureau of Standards has carried out tests, right, sponsored by the Navy's Bureau of Aeronautics. General conclusions that may be drawn from these investigations include:

Reinforcements are more effective in reducing the stress concentration at a hole when they are fastened with at least two rows of rivets or bonded to the sheet so that reinforcement and sheet act as an integral unit. When attached in this way, the plane stress theory can be used in the elastic region for computations of the average median surface stresses and of changes in hole diameter. Reinforcement on one side of the sheet only introduces bending moments tending to bring the reinforcement into the plane of the sheet and may introduce bending stresses in the reinforcement of the same order of magnitude as the median fiber stresses. A cutout in a square panel, particularly if the panel is under shearing load, may reduce the buckling stress considerably. Reinforcement of the hole may increase the compressive and shear buckling stresses to values higher than those for a panel without a hole.

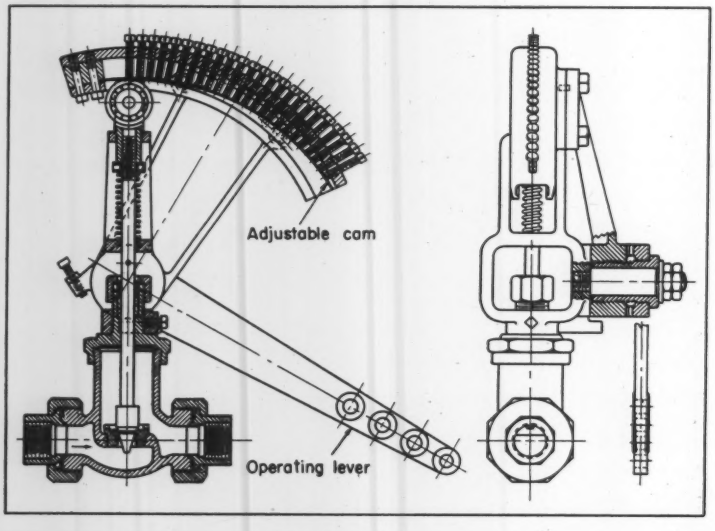


Assembly of plastic parts by machine, employing principles of metal parts assembly, produces all-plastic styrene harmonicas at the rate of one each second in the plant of the Magnus Harmonica Corp. Five plastic parts—top cover, top reed plate, comb, bottom reed plate, and bottom cover—are loaded manually

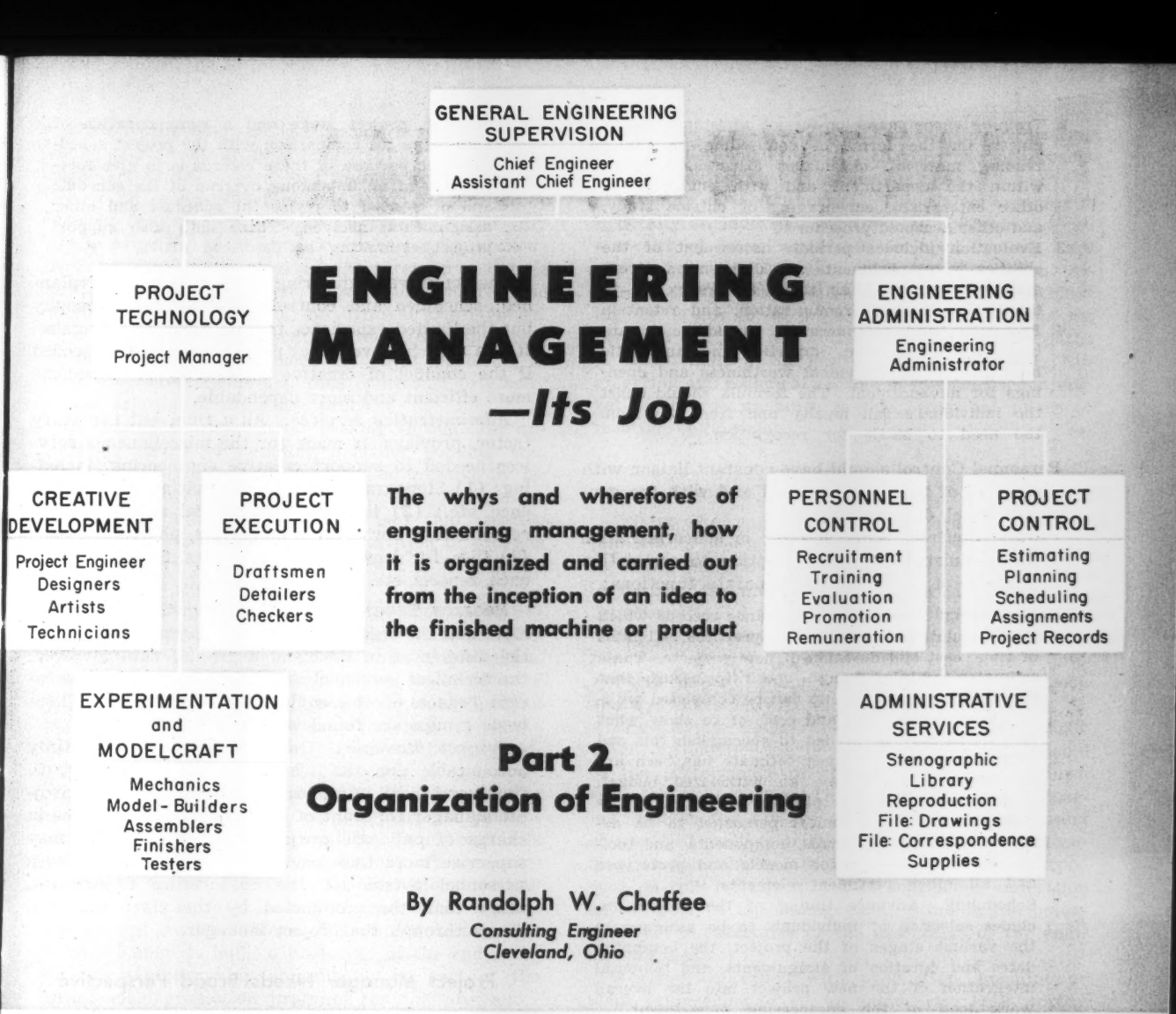
into hoppers. As the assembly line rotates, the parts are fed in correct order by gravity into fixtures. Adjoining each hopper is an applicator through which adhesive is fed from a central well. Pressure is applied to clamp the parts together, building up the product as it moves around the circle. At the end of the cycle each harmonica is ejected from the machine, above, and an electronic device automatically inspects for size and shape. These five molded plastic parts replace 80 machined parts. Also, a pretuned reed plate replaces laborious hand riveting and hand tuning operations previously employed.



Automatic gaging unit, above, on a Besly grinder measures thrust washers as they leave the machine and applies the results to actuate the infeed of the grinding wheels, insuring production within allowable tolerance limits. The gage employs two tungsten carbide anvils, one fixed and the other reed-mounted. Movement of the latter actuates an air jet connected with a Sheffield gaging head. Air flow above or below the gage settings, caused by oversize or undersize parts, produces electrical impulses in the unit. Transmitted through a control box to solenoids, these impulses energize the feeding mechanism of the grinding wheels. Depending on the conditions of grinding, the electrical system can be set to energize movement of both wheels simultaneously or each wheel alternately as may be desired by the operator.



Adjustable cam on the valve at left provides for setting the valve lift at the required position within the range of the valve movement. Thus flow ratio between two variables may be controlled for all positions of the valve. Designed by Atlas Valve Co., the valve has a flexible cam track attached at one end to a cam arm which is rotated by the operating lever. Curvature of the cam is adjusted by screws to give the profile desired. A roller attached to the valve stem is held in contact with the cam by a spring on the stem. In this way backlash on the lever movement is minimized because the roller is always in contact with the cam for both directions of travel.



By Randolph W. Chaffee
Consulting Engineer
Cleveland, Ohio

NO TWO engineering projects are alike and different engineering organizations can bear only a faint resemblance to each other. From the standpoint of basic principle, however, there are two structure patterns which should appear in any engineering organization and each of these patterns reflects differences in the kinds and degrees of skill employed.

ADMINISTRATION OF ENGINEERING: It is not yet widely recognized that administration is a necessary and distinct part of engineering. Where administration is neglected, technology is unable to bring the full weight of its talents to bear upon its tasks. Where technical personnel are compelled to carry the administrative burden, not only does technology suffer from neglect and delay, but also the administration is indifferent, inefficient and inaccurate.

Engineering administration should be recognized as a separate function apart from engineering technology, although it may be the job of but one person or it may require the services of many. It is important mainly to treat administration as an important ally of engineering technology.

Engineering Administrator: Under general direction of the chief engineer, the engineering administrator is responsible for all interests of the department other than engineering technology, for liaison with other departments at his level and for all services which support and aid the technical engineering staff of the enterprise.

Personnel Control: Because personnel administration is a specialized art, provision is made for skilled handling of the engineering personnel. Although some of this work may be done in the personnel department of the enterprise, much of the work can be done intelligently only within the engineering department. Some of the activities included in this section are as follows:

1. Recruitment includes the seeking out and screening of prospective members for the engineering staff, with actual placement subject to approval by superior project technologists

2. Training encompasses on-the-job assistance to engineers in the forms of counselling, providing reading material, organizing discussion groups within the department and with engineers of other enterprises, encouraging of outside study, and other accepted procedures
3. Evaluation includes periodic assessment of the abilities, accomplishments and deficiencies of engineering personnel as these factors relate to training, promotion, remuneration, and retention
4. Promotion and remuneration should be organized on an objective, impartial and automatic basis subject to individual worthiness and openings for advancement. The formula should enlist the individual's full loyalty and free him from the need to battle for recognition.

Personnel Control should have constant liaison with all members of the technical staff and with the general personnel department.

Project Control: This activity is concerned with project administration from inception to completion and consists of the following separate functions:

1. Estimating. Using the performance records which it accumulates, project control prepares estimates of time, cost and duration of new projects. These estimates contain enough detail to assure that the new project as set up can be completed within the allowable time and cost, or to show what alterations may be needed to accomplish this end
2. Planning. After a project estimate has been approved and has become an authorized budget, project performance is planned in respect to its work stages, the technical personnel to be assigned to it, the materials, components, and tooling to be purchased for models and prototypes and all other pertiment elements
3. Scheduling. Advance timing of the project includes selection of individuals to be assigned to the various stages of the project, the beginning dates and duration of assignments, and temporal integration of the new project into the overall work load of the engineering department
4. Assignments. Individuals are given specific project assignments with clear instructions as to working material, allowed time, outline of work to be completed, scope of responsibility, etc.
5. Project Records. These include primarily the record of man-hours expended by each individual on each project stage and a summarization of man-hours for comparison with the project schedule. The purpose of these records is to give forewarning of an impending overrun of the schedule and of a need to revise the schedule and other assignments involved. These data also support project estimating as described above.

That creative engineering endeavor can be preplanned, scheduled and controlled is a new philosophy but the limited experience to date gives high promise. It is a fact, however, that proper procedure is needed if the conduct of creative engineering is to become more efficient and more dependable.

Administrative services: As a final but necessary factor, provision is made for the miscellaneous services needed to support creative engineering, including: (1) Stenographic, for report-writing, correspondence, etc.; (2) library, for technical reference; (3) reproduction, for making blueprints, photostats, etc.; (4) files, for drawings; and (5) files, for correspondence, reports, etc.

PROJECT TECHNOLOGY: Following the basic principle that the organization of engineering must reflect the differences in kind and degree of skill involved, the technical personnel are placed in a specific separate branch of the engineering department. Three basic groups are found within this branch.

Project Manager: One individual is held fully accountable for the whole project from start to finish and in all of its ramifications. Called the project manager for want of a better term, he may be in charge of only one project at one time or he may supervise more than one. All technical liaison with personnel outside of the engineering department, other than that conducted by the chief engineer, clears through the project manager.

Project Manager Needs Broad Perspective

The project manager may not necessarily be an engineer of supreme talents since he is concerned as much with the broad overall objectives and limitations of the project as he is with its intimate technology, but he must have a good general working knowledge of all of the technical arts involved.

". . . seeking out and screening of prospective members for the engineering staff . . ."

Photo, courtesy Goodyear Tire & Rubber Co.

Creative Development: All technical personnel who make significant original contributions to the technical art of the project are in this group and represent all of the different arts involved. The group may be headed by a mechanical engineer if the project is primarily mechanical in character, by a chemical engineer if chemical in character, etc., but as project engineer he must be able to interpret and adapt to the project all of the different arts involved.

Also included in the creative group are junior technicians who work out the component problems in their respective fields and thus contribute originally to the solution of the whole problem.

Project Execution: After passing through the apprentice stages of detailing drafting, copying etc., the young engineer may prove to be creative by instinct and move on into the creative side of engineering. On the other hand, he may lack that creative talent but have an instinct for execution of projects of a high and valuable order, in which case he appears in this group.

The responsibility of project execution is to express on paper, accurately and clearly, the creative concepts developed. This group includes draftsmen, detailers, checkers and others. They make little or no creative contribution of their own to the project beyond depicting it for the information of others who will fabricate, assemble and deliver the end product of the creative engineering function.

Experimentation and Modelcraft: This is primarily a shop function and the responsibility of persons manually skilled in forming and fashioning materials into articles, objects, mechanisms, etc. The primary purpose is to set up models for operational testing to prove or disprove assumptions made during the preliminary creative efforts. This function may also be called upon to build a prototype of the end product for photographing and display as part of the initial sales promotion program for the product.

The foregoing discussion of functions does not attempt to reflect all of the possible components of an engineering organization for this would be impractical in an article of this scope. What is sought here is proper emphasis upon two basic principles, namely: (1) Administration is a separate and distinct part of the engineering organization, and (2) the engineering organization should reflect sharply the differences in kind and degree of skill employed.

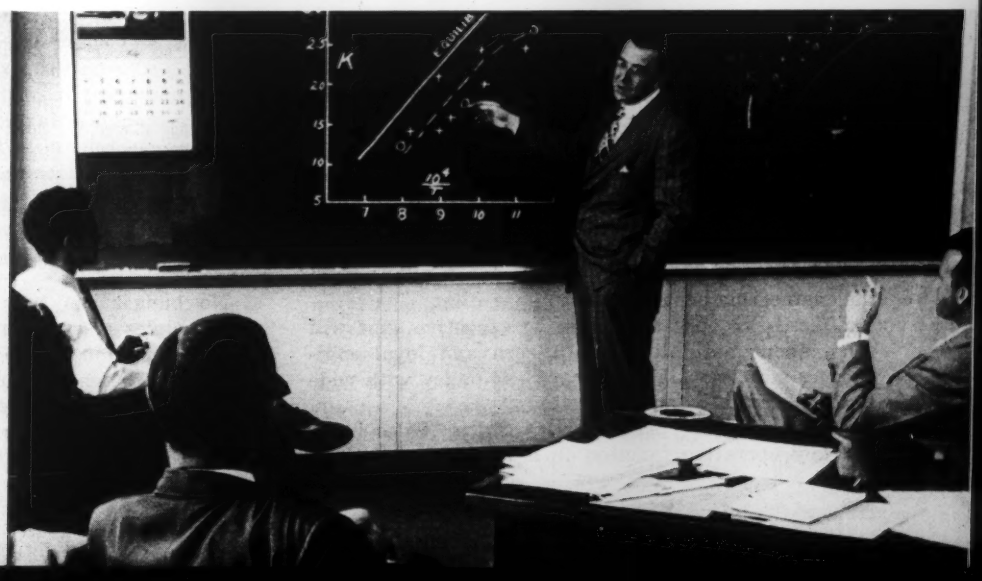
CONDUCT OF ENGINEERING: The typical engineer is generally an individual who has at his beck and call a wealth of technical talents devloped by long hard years of personal study, training and application. Although highly practical in his technological thinking, he is apt to be subjective and emotional in his personal behavior and be led, thereby, into the paths of isolationism in his professional environment. Engineering and technical schools in their curricula have yet to include guidance in this important phase of creative engineering.

Right Professional Attitude Necessary

All too often engineering staffs regard themselves as inhabitants of a special, exalted and detached domain to which must come on bended knee the corporate executives, the manufacturing department, the sales department, and the other major branches of the enterprise. However comfortable this point-of-view may be, it hardly endears the engineer to his fellow-workers or enables him to achieve that greater good which evolves only from active co-operation.

Engineering Co-operation with Executive Staff: Often, an engineering department may be restricted to the performance of specific tasks handed down from top management. These might comprise preparing designs for pricing of products, the designing of specified tooling, or discharging other minor technical responsibilities. In most cases, a limited scope such as this reflects a lack of confidence and vision on the part of top management, either in the potentials of the enterprise or the usefulness of the engineering staff.

Whether concerned with production, machine or plant engineering or all, an engineering staff worthy of the name will not only execute smartly the tasks assigned to it by management but also will originate sound and practical projects for general betterment of the enterprise. These projects are preplanned in all essential detail and presented for the approval of management in such form that management has



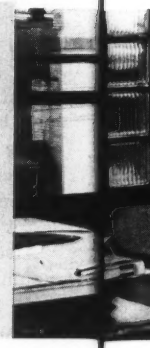
". . . project manager . . . is concerned with the broad overall objectives and limitations . . ."

Photo, courtesy Standard Oil of New Jersey

Photo, courtesy General Electric Co.

". . . models for . . . testing . . . assumptions made during preliminary creative efforts . . ."

". . . engineering is best qualified to conceive, try and recommend new . . . uses which increase saleability . . ."



Photo, courtesy American Brake Shoe Co.

only to reply "yes" or "no" to the proposal.

No one likes a braggart or suggests that any engineering staff blow its own "horn" too loudly or too often. On the other hand, it does the engineering department or its employing enterprise little good to keep entirely mum about engineering activities and accomplishments. Managements should and do like to know occasionally what they are getting in return for engineering expense. Any engineering department should report regularly, briefly and pointedly to management concerning its program, problems, accomplishments, and plans.

Co-operation between management and engineering is a two-way affair and management should co-operate with engineering by calling the latter in for joint consideration of mutual problems, policies and plans wherever engineering can contribute—and this is nearly always. It would be futile for management to reach a decision to add a product line when engineering can point out certain good reasons why that product cannot be engineered, designed or manufactured to advantage.

Engineering should keep constantly abreast of new technological developments in materials, processes, methods, etc., and strive to keep its product modern by the uses of such new art in the right places and at the right times. The initiative for this forward-looking program can best be taken only within the engineering department.

Effective co-operation with the executive staff will come about naturally with a keen and loyal engineering staff working under a clear policy with wide latitude for independent action.

Engineering Co-operation with Sales: Sales being the key operative function in a normal competitive industrial environment, engineering's co-operation with this function is second in importance only to co-operation with top management.

Apart from giving sales an easily saleable product as hereinbefore described, engineering can co-operate with sales in many ways. Newer products that are engineered or designed for each sales transaction become standardized as their applications increase, and the time comes when these products can be cataloged, priced and stocked as standard items for best service to customers. The sales department is less likely to discover the arrival of that time than the engineering department, and the latter should take the initiative in recommending the standardization program.

The engineering department is best qualified to conceive, try and recommend new product uses which increase saleability, and should pass these suggestions to the sales personnel. Also, the engineering department is an excellent training ground for salesmen and can help the sales force to become more effective through training in the engineering operations. In addition to participating in the setting up of analytical programs by advertising and market research groups, engineering can further co-operate in sales promotion by providing information for advertising and display copy, and application data accumulated during the original development of the product.

Many engineering departments are subject to frequent or continuous pressure from the sales force to adjust schedules in order to give preference to favored customers. Sales people are opportunists in large measure, motivated by circumstances of the moment, and many cannot resist the impulse to close a large order on the strength of early delivery. They may easily fail to realize the effect of a major upset in the engineering schedule upon services to other previous customers, and if the tendency is tolerated too far, the engineering department's schedule may be completely and irrevocably ruined.

Limited Alterations and Changes

No budget can be utterly inflexible; alterations in schedules must occur and must be accepted. However, there must also be some limit to schedule changes. The engineering department should co-operate with the sales force by maintaining a composite, easily understood record or chart of its schedule to show the effects of any changes. If the sales force

can be shown these effects before changes occur and be asked to assume the responsibility for them before their customers, demands for schedule changes diminish to a reasonable minimum.

Engineering Co-operation with Manufacturing: Among the many claimants for engineering co-operation is the manufacturing function which can make a great many diverse demands for interpretation of the engineered designs of product, tooling, materials, and methods. Usually, a demand for help from the manufacturing department comes after the latter has started something, has gotten into trouble and must hold up everything until engineering can straighten things out. Engineering must be prepared to give aid to manufacturing at any time and at a moment's notice. Where the engineering staff is large enough, one or more members may be designated for liaison with manufacturing to assure prompt service to the latter without upsetting the engineering schedule. If several such liaison men are provided, one may concentrate upon product, another upon tooling, a third upon materials, etc.

Design for Production Imperative

It is most important that the end products of engineering be manufactureable within the policies and methods of the manufacturing organization, in least time and at least cost. Engineering co-operates with manufacturing, therefore, by keeping fully informed of manufacturing capabilities and limitations and by keeping its engineering technology reasonably confined within those capabilities and limitations.

Perhaps the most frequent liaison at the lower engineering levels will be with opposites in manufacturing and will involve every conceivable subject and problem. It may seem at times that the engineering function is but a small and subordinate element of manufacturing. Although this situation may prove irksome to engineering, it is nonetheless necessary and should be accepted in good grace.

Engineering Co-operation with Finance: Engineering co-operates with the finance function primarily in the preparation of annual engineering budget estimates and by sincere effort to confine expenditures within the authorized budget limits.

Preparation of budget estimates requires a careful preview of the coming year's program, an evaluation of the relative effect of many potential projects upon the overall economy of the enterprise, a careful selection of the more promising projects, a costing of those projects, and a general plan for the accomplishments of the department. In particular, engineering should be prepared intelligently to resist any effort of management to set up a fiscal budget based solely or largely upon last year's budget or expenditures. Circumstances may dictate either an increase or a decrease from last year's budget and only engineering can supply complete evidence in support of either.

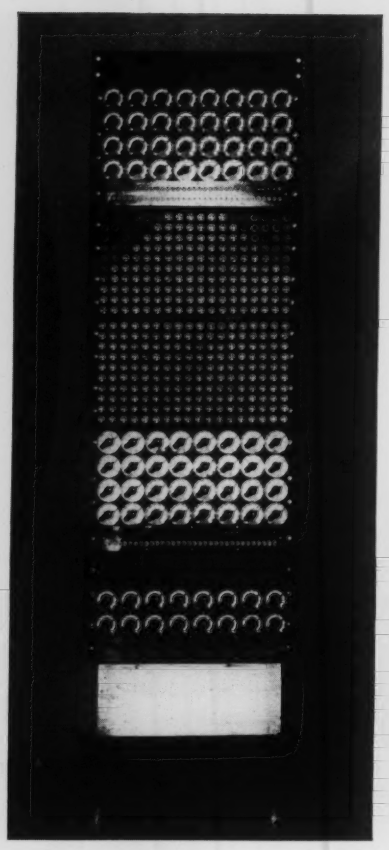
The process of living within a budget requires the maintenance of accurate records of expenditure by project, stage of work and object of expenditure, and a periodic comparison of actual expenditures with the current budget. Engineering should forewarn finance when a budget overrun impends and present logical evidence to support a request for budget revision.

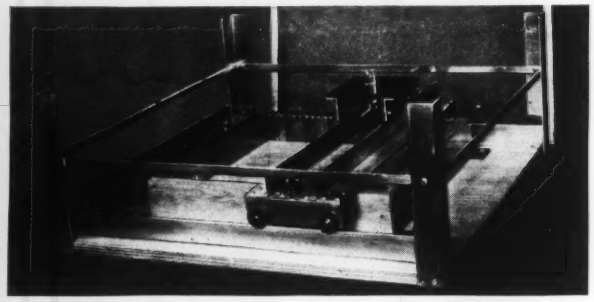
Engineering may further aid finance by preparing statements of requirements for materials which will be used by other departments of the enterprise, but which are best known to engineering. Typical of this are materials to be used by manufacturing during the coming year on projects currently under development in the engineering department.

Engineering Co-operation with Purchasing: Engineering can be of great value to purchasing in supplying specifications of purchased raw materials and component parts, special instructions for packaging, shipping and handling, unit costs as used in securing budget and project authorizations, and other information which will aid purchasing to meet time schedules and cost limitations. This assistance to purchasing should flow naturally out of the engineering operations of preparing bills of material, design specification data, performance data, etc.

GENERAL SUMMARY: It may appear, at this point, that engineering must regard itself as the slave of all other branches of the enterprise and the master of not even its own destiny. Whether or not this is actually true, it is a healthy attitude for every engineer to take and the one most likely to advance the interests of the enterprise, the engineering profession and the engineer himself.

However lofty his position, the individual professional is, in fact, a personal servant to all his fellowmen and must remain dedicated to personal service to others at whatever cost to himself. No engineer or other professional has achieved enduring greatness by following a lesser philosophy.





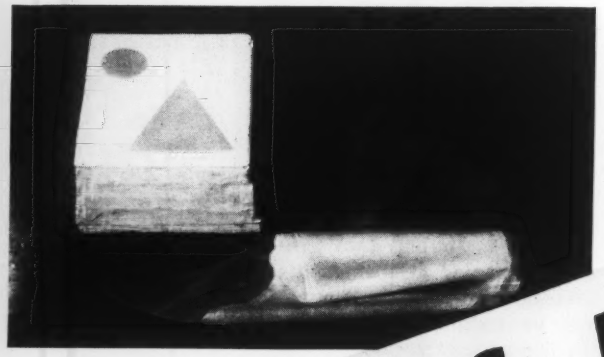


Fig. 1—Typical equipment for torsion analysis showing a plate with a triangular hole and a circular calibrating hole for the membrane analogy, an electrolytic tank for conducting sheet models, and a resistance network panel for solving torsion problems

BARS IN T

. . . methods for analyzing torsio

THE bar in torsion is an important element in many mechanisms and drive systems. When the bar cross section is irregular, as is the case when keyways or splines are involved, or when structural shapes are used, the determination of optimum proportions from the standpoint of strength, fatigue and rigidity becomes a real problem. As the situation now stands, the designer lacks a generalized perspective of the many possible methods by which such problems might be analyzed. It is the purpose of this article to provide such an overall perspective. With this in mind, a designer would be in a position to adopt a method of analysis best suited to his particular problem or range of problems. Typical equipment for three of the methods discussed is shown in *Fig.* 1.

For most regular sections the designer may refer to handbooks for torsional stress and rigidity. More often, the section is of such form that handbook values only provide an indication of the general order of stress without showing conclusively how an optimum shape should be designed. The designer must then resort to a fundamental analysis of the problem.

NUMERICAL SOLUTION: Theoretical solutions of the torsional problem have been accomplished for regular symmetrical sections such as the ellipse, the square and the equilateral triangle, giving the stress concentrations and rigidities in terms of the cross section dimensions[1]. Such solutions are important in showing that the torsional shear stress is greatest on the boundary and occurs at those boundary points which are closest to the center of gravity of the cross section. However, this does not ordinarily solve the immediate problem of the designer.

[1] References are listed at end of article.

By Walter W. Soroka
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SION

ar stress

An approximate numerical solution is always possible for any cross section. This method requires a minimum of special equipment and is undoubtedly the best approach when only an occasional problem is encountered. It suffers from the disadvantage of being tedious and long for complicated sections. A considerable speeding up of the process is possible if a punched-card system is available for performing the numerical calculations automatically[2,3]. Since punched-card machines are often available in the accounting divisions of manufacturing and engineering organizations, it is frequently possible to work engineering problems into the schedule if the accounting division equipment includes a calculating punch of the type 602, 602-A or 604. It should be mentioned in this connection that the International Business Machines Corp. has issued very recently a preliminary manual of information on the Card-Programmed Electronic Calculator developed specifically for engineering calculations rather than for accounting purposes. This calculator is composed primarily of standard machines interconnected so as to solve engineering problems.

To illustrate the basic method of the numerical procedure, consider a bar having a cross section of the form illustrated in *Fig.* 2. The section is overlaid with a grid of dotted lines forming small squares over the entire cross section. At each intersection point of the type illustrated in *Fig.* 3 the stress function ϕ from which the shear stress is to be determined must satisfy the equation

$$\phi_o = \frac{\phi_1 + \phi_2 + \phi_3 + \phi_4 + 2h^2G\theta}{4} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where h is the network spacing, G is the shear modulus of the bar material and θ is the applied twist per unit length of bar.

The objective in the numerical procedure is to choose the values of ϕ_1, ϕ_2, ϕ_3 and ϕ_4 in such a manner for all points in the interior of the cross section so that Equation 1 is satisfied at every point.

For points near curved boundaries, two alternatives are possible. In one, the curve is approximated by a jagged line in such a manner as to maintain complete squares throughout the network overlay. If the squares are taken small enough, the jagged line approximation may be very close. A finer net may be used near the curved boundaries than that used over less critical regions. In any case, if this alternative is used, Equation 1 is the only equation which needs to be satisfied. It can be shown that the true values of ϕ around the boundary are all zero. Thus, by assigning the value zero to all network points representing the boundary of the cross section and by assigning any arbitrary values to all interior points, the values at the interior points may be corrected one by one by means of Equation 1. By starting with the first interior point on the left in the top row, for example, correcting its arbitrarily-assigned value by means of Equation 1, then going on to the next point in the same row and correcting it, repeating the procedure from point to point and going from row to row (remembering, of course, to leave the boundary values unchanged at zero) a first iteration cycle will have been completed when the last interior point has been reached and corrected. The values so obtained will be nearer the truth than those which had been arbitrarily assigned to begin with. By performing a second iteration, still closer results will be obtained. After a number of cycles (depending on the accuracy required and the complexity of the problem) stationary values of ϕ will have been obtained at the network intersection points. The repetitive nature of this process makes it ideally suited to automatic machinery of the punched-card type.

The second alternative for points near curved boundaries is to take the actual intersection points

***TECHNIQUES** whereby a designer may study torsional shear stress distributions in uniform bars of arbitrary cross section are discussed in this article. Included are the numerical iteration procedure, electrical resistance networks, electrically-conducting sheets, membranes, the hydrodynamic simulator, and a three-dimensional photoelastic stress freezing method*

of the overlay with the cross section boundary and to modify the improvement formula to account for the shortened lines between intersection points. *Fig.* 4 represents a typical condition near a curved boundary, the spacings ah and bh being less than h (a and b are numbers less than 1). For this case the improvement formula becomes

$$\phi_o = \frac{ab}{a+b+2ab}\left[\frac{\phi_1}{a} + \phi_2 + \phi_3 + \frac{\phi_4}{b} + G\theta h^2\left(1+\frac{1}{2}a+\frac{1}{2}b\right)\right] \quad \ldots\ldots (2)$$

This method thus requires a change in formula for points near curved boundaries and so is more difficult to apply. However, the boundary conditions are more faithfully reproduced under these circumstances so that a coarser net may be used at the boundary than that required for the first alternative.

The shear stress at any point on the boundary of the cross section is given by the gradient of ϕ in a direction perpendicular to the boundary at the given point. Thus, in *Fig.* 2, the shear stresses at points A and B on the boundary are

$$S_A = \frac{\phi_m}{h}\,;\; S_B = \frac{\phi_n}{(6-4\sqrt{2})h} \quad \ldots\ldots (3)$$

ELECTRICAL NETWORKS: If an intensive study is to be made of a given problem or if a wide variety of problems is to be solved, an experimental technique may well turn out to be a most economical procedure. The numerical iterations which are laboriously done on desk calculators may be accomplished automatically and instantaneously by current flow in an electrical resistance network such as illustrated in *Fig.* 1. Thus, once the network is connected to a current source the values of ϕ may be read immediately by means of a voltmeter at the intersection points of the overlay. By quickly switching from point to point around the first inner contour of intersection points near the boundary, the points of maximum ϕ (i.e., of maximum voltage) are rapidly determined.

Construction Is Simple

The resistance network is easily constructed; all that is required is the replacement of each line of length h in the overlay of *Fig.* 2 by a resistor of magnitude R. If foreshortened lines are used at curved boundaries, the corresponding resistors are of proportionately reduced magnitude aR or bR. The resulting network is shown in *Fig.* 5. In the resistance network, *Fig.* 1, the knobs are for the variable boundary resistors. The central panels contain plugs which, when inserted, connect four adjacent resistors to form a nodal point.

In addition to the decision which must be made regarding the shape of the boundary (i.e., whether to use complete squares and a jagged boundary, or to use foreshortened sides so that nodes will fall exactly on the boundary), the designer has two alternatives in setting up voltages at the boundary. In the first and perhaps more direct method the boundary voltage is set equal to zero (corresponding to $\phi = 0$) and a current of magnitude $2h^2G\theta/R$ is fed into each node. For those nodes involving reduced resistors the current input is adjusted to $G\theta h^2(1+\frac{1}{2}a+\frac{1}{2}b)/R$. The individual adjustments of these currents may be rather inconvenient. These may be accomplished easily and simultaneously if a large resistor, of the order of $100R$ to $1000R$, is connected to each network node, the other ends of these resistors being connected to a battery of sufficiently high potential to give the required current. Variations of voltage over the network would then have only a minor effect on the magnitude of this current and no painstaking adjustments of rheostats need be made. This is the method adopted by Redshaw[4] in his potential analyzer. The voltages at the network nodes then give the stress function ϕ directly.

Network Is Halved if Symmetrical

The other alternative avoids the need for feeding current into the network nodes. Instead, the boundary voltages are set so that they vary from point to point in accordance with the equation

$$\Phi_b = \frac{G\theta}{2}(x_b^2 + y_b^2) \quad \ldots\ldots (4)$$

where the subscript b indicates that the values are only for the boundary. The origin of co-ordinates x and y may be taken anywhere; however, if the co-ordinate axes are so directed as to coincide with axes of symmetry of the cross section, if any, a simplification in the setup is possible in that the network need represent only the material on one side of an axis of symmetry. Thus, in *Fig.* 5 there is actually represented twice as much network as is absolutely necessary for a solution of the problem. The same is true of the numerical overlay of *Fig.* 2. An axis of symmetry acts like an impenetrable wall of insulation, so that the network intersection points along this axis would remain disconnected.

If the voltage distribution over the boundary of the network is taken in accordance with equation 4, the stress function ϕ of equation 3 is given in terms of the measured voltage Φ at interior network points by the equation

$$\phi = \Phi - \frac{G\theta}{2}(x^2 + y^2) \quad \ldots\ldots (5)$$

where x and y are the co-ordinates of the interior points. Thus, Φ is determined by voltage measurements at the network nodes; then ϕ is calculated by means of equation 5. The shear stress at the boundary then follows from equation 3.

ELECTRICALLY CONDUCTING SHEETS: The error introduced by the use of a finite net of spacing h in either the numerical or electrical methods of analysis is reduced with a reduction in net spacing. Such a reduction in spacing greatly increases the labor of calculation or the labor and cost of setting up a resistance network. However, it is possible to go to a net of zero spacing in the electrical analog without much added inconvenience by using a thin conduct-

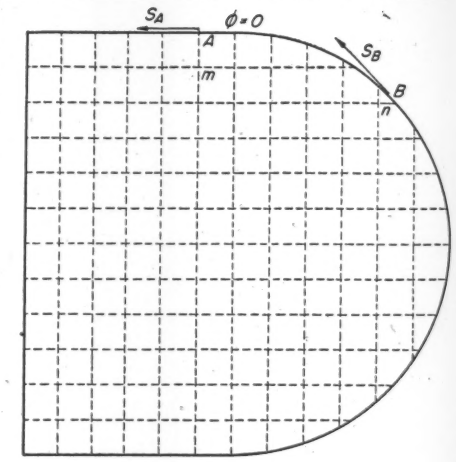

Fig. 2—Left—Bar cross section with square network overlay of side h for numerical torsional analysis

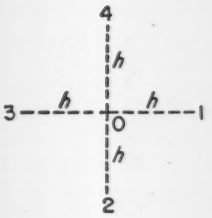

Fig. 3—Right—Typical interior network point

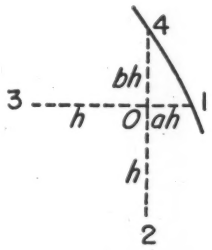

Fig. 4—Left—Typical network point at curved boundary

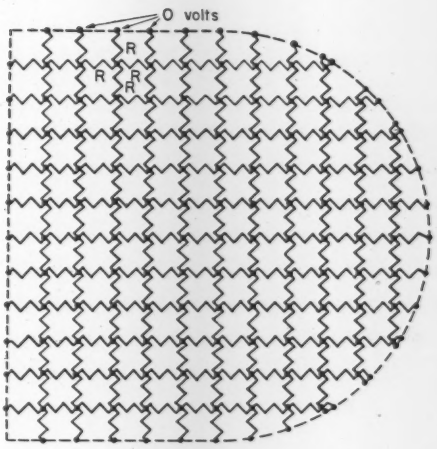

Fig. 5—Right—Electrical resistance network for analog solution of torsional problem

ing sheet to represent the bar cross section. This sheet may be of metal foil, of conducting plastic, of metallized paper or it may be a thin layer of electrolyte, usually tap water. If solid conductors are used, direct voltages and currents may be applied; if electrolytes are used, alternating voltages and currents are necessary because of the polarization and electrolysis occurring with direct-current supply.

The conducting sheet may be formed to represent the cross section of the bar exactly to some enlarged scale. However, some approximation is entailed in applying currents or boundary voltages. Thus, in the first alternative described above, a boundary voltage of zero may be applied very closely. But it is necessary to introduce a current of uniform density $2G\theta/r$ (where r is the resistance per unit length of a unit width of conducting sheet) per unit area over the entire surface of the sheet. This is not at all practical from an engineering standpoint. Consequently, the uniform distribution is approximated by feeding current into a discrete number of points situated over the surface of the conducting sheet, such that the total current flow divided by the area of the sheet equals the desired uniform density. In order to minimize the effects of local high current densities, the voltages are read by means of probes placed intermediate between current electrode locations.

Since the conducting sheet is continuous, it is possible to measure voltage gradients close to the boundary by close spacing of the measuring probe.

In the second alternative, the problem of feeding current into the surface of the conducting sheet does not arise. However, there is the problem of obtaining the voltage distribution of equation 4 along the edge of the sheet. A continuous distribution is not attainable practically; instead, a step-wise distribution is used, which approximates the continuous distribution more and more closely as the number of steps is increased. The steps are obtained by attaching narrow electrodes along the edge of the sheet, *Fig.* 1, and applying a different voltage to each electrode in accordance with its position with respect to the origin of co-ordinates, Equation 4.

MEMBRANE ANALOGY: For designers who are not on sure ground when electrical circuits are involved, there are a number of purely mechanical methods which will produce substantially the same results as the electrical networks. Noteworthy among these is the tried and proved "soap film" or "membrane" analogy developed to a skillful degree by Griffith and Taylor[5].

As in the electrical counterpart, two alternatives are possible. In one, a soap film is stretched over a hole in a flat sheet of metal, the hole having the same shape as the cross section of the bar in torsion. By clamping this sheet over an airtight box, the film may be subjected to a slight air pressure which causes

it to bulge upward measureably. The vertical positions of points on this film with respect to the flat sheet may be denoted by ϕ, since they correspond to the stress function of the numerical iterative procedure and to the voltages measured in the conducting sheet with current input. Since the gradient of ϕ normal to the boundary determines the shear stress, it is necessary only to measure the slope of the soap film at the edge of the hole by optical or other means. A direct calibration of the soap film is attained by also stretching the film over a circular hole cut in the sheet next to the test hole. The relative slopes of the two films then give the relative shear stresses in the two bars of the same material and subjected to the same twist per unit length.

In the second alternative, the use of air pressure is eliminated by distorting the edge of the hole in such a manner that its elevation, ϕ_b, measured from a horizontal reference plane is given by Equation 4. Equation 5 may then be used to calculate ϕ and the gradient of ϕ normal to the boundary, i.e., the shear stress. Some workers[6] have used thin uniformly-stretched rubber sheets instead of soap films. In *Fig.* 1, for instance, a dental dam latex rubber sheet was used as a membrane. The pressure was that supplied by the hydrostatic head of a wet plaster mix which hardened into the block of plaster shown. Having a solid plaster surface on which to work facilitates measuring stress contours and preserves the membrane sheet for future reference.

Color Streaks Show Stress

HYDRODYNAMIC ANALOGY: For designers with a flair for hydrodynamics, there has been disclosed[7] recently a new technique developed at the University of Michigan for mapping fluid flow fields, which may be used to solve the torsion problem. The method consists of building up a uniform bed of finely granulated material (copper shot), the plan form of the bed being that of the cross section of the given bar. A glass plate spaced slightly above the bed provides a thin horizontal space for fluid flow to occur. Water under a slight hydrostatic head oozes up through the bed and then flows out horizontally under the glass plate. Fine crystals of coloring agent on the bed produce streaks which are readily photographed through the glass plate. A graphical construction of the perpendicular family of lines gives the lines of constant stress function from which the shear stress may be calculated.

PHOTOELASTIC TECHNIQUE: The recent introduction of a new photoelastic resin, Fosterite[8] for use in stress-freezing techniques has provided a more direct means for the analysis of stresses in bars of arbitrary cross section. Fosterite is available in heavy rods several inches in diameter and several feet long so that relatively large test bars may be machined. By loading such a bar under twist in an oven whose temperature is gradually raised to the first critical point of the material and then gradually reduced to room temperature, the load may then be removed but the shear stresses will remain frozen in the specimen. Thin slices of the material may be cut and examined in a polariscope for shear stress distribution. For calibration purposes a circular bar specimen subjected to a known torsional couple should be carried through the oven simultaneously with the test bar and similarly examined in the polariscope for shear stress distribution.

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Carmichael Edits Kent's Handbook

E. L. Shaner (left), chairman of the board of the Penton Publishing Co. and editor-in-chief of *Steel*, and Colin Carmichael, editor of MACHINE DESIGN, are shown with Edward P. Hamilton, president of John Wiley & Sons, celebrating the publication of the new twelfth edition of *Kent's Mechanical Engineers' Handbook*. They are looking over a copy of the *Design and Production* volume of the handbook, edited by Mr. Carmichael.

The reception honoring Mr. Carmichael and J. K. Salisbury, division engineer with the General Electric Co. who supervised the preparation of the *Power* volume of the handbook, was held recently at the

Van Curler Hotel, Schenectady. Attending the celebration were contributors to the handbook, GE officials, and several other prominent scientists and engineers.

First published by Wiley in 1895, the Kent handbook represents the work of 172 leading engineering specialists and was in preparation for the past four years. The current edition represents a 20 per cent enlargement over the preceding edition.

Production Case Hardener

By W. L. Corteggiano
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BATCH methods of heat treating are apt to be the bottlenecks in many modern high-speed production lines. With the increasing application of radio-frequency induction heating equipment to metal-working production lines, new work handling systems are needed to extend the utility of this tool for continuous, high-speed production lines.

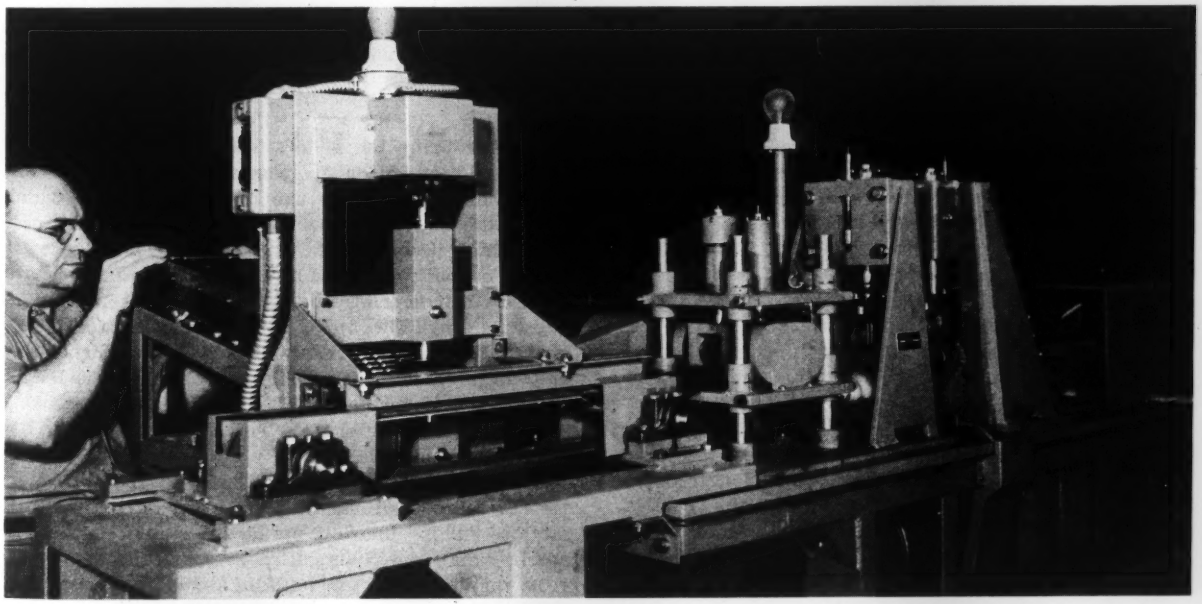
A Westinghouse development, the horizontal rotating scanner with automatic work loader has been designed to selectively case-harden miscellaneous cylindrical parts. It was found necessary to limit the scope of the machine to a specific range of shaft sizes, although maximum flexibility is desirable. As designed, it will handle shafts from $\frac{3}{8}$ to 2 inches in diameter, and from $1\frac{3}{4}$ to 18 inches in length. This range represents, for example, a cross section of the various size shafts used in the automotive industry, such as wrist pins, shifter rails, clutch shafts, tractor bushings, guide posts and other miscellaneous parts.

Shown at the left of *Fig.* 1, the loader starts the workpieces on their way for processing. It is equipped to receive several different types of magazine loaders so designed that the operator cannot run anything except those parts for which the machine is adjusted, i.e., he cannot rotate the workpieces longitudinally, producing an incorrect hardness pattern.

The pieces roll down the magazine and are released to the conveyor belt one at a time in synchronism with those leaving the scanner. Two solenoid-operated selectors operate, first to allow one piece to roll onto the conveyor and then to let another drop into the ready positon. The conveyor, shown in *Fig.* 2, has guides mounted above it to position the parts directly in line with the pick-up rolls on the feeder unit. The conveyor belt moves slightly faster than the train of pieces in the scanner, maintaining a continuous flow of workpieces.

Shown at the left of *Fig.* 3, the feeder unit picks

Fig. 1—Case hardening small cylindrical parts by radio frequency induction heating on the Westinghouse horizontal rotating scanner. Once started, machine is fully automatic



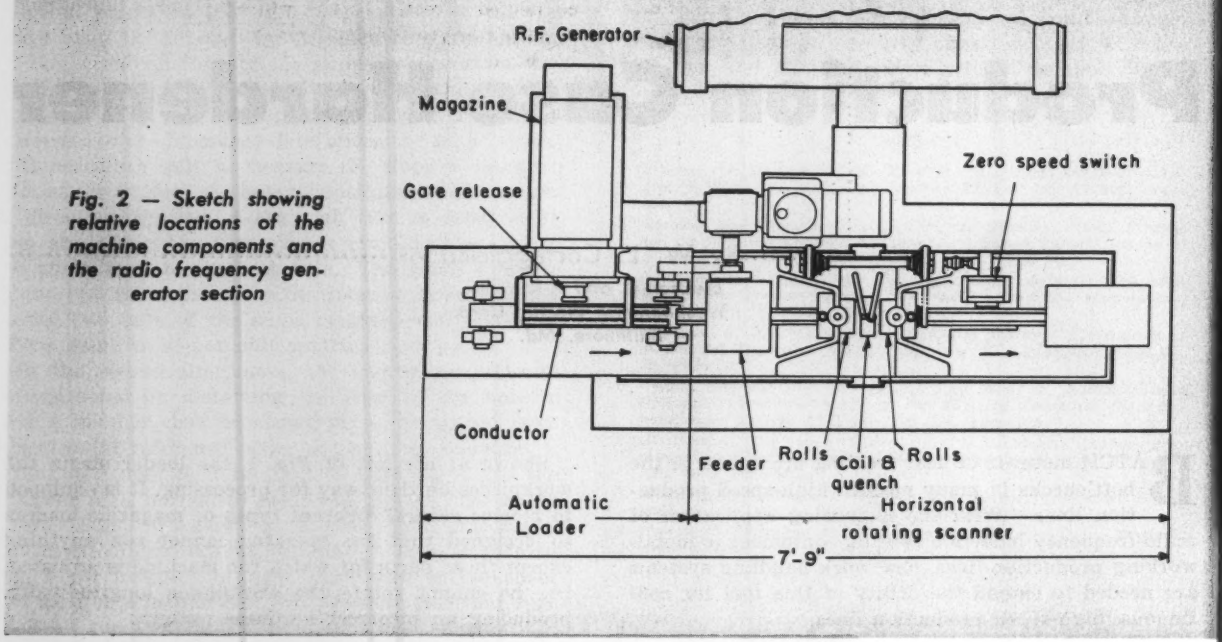


Fig. 2 — Sketch showing relative locations of the machine components and the radio frequency generator section

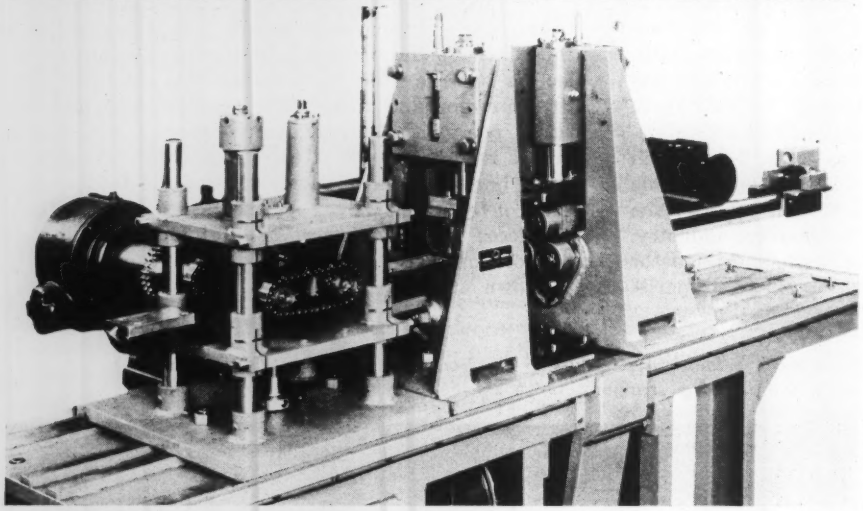


Fig. 3—Partially disassembled view of the scanner alone. Feeder unit at left moves stock to heating section, center, then under zero speed switch wheel and to discharge end of machine at right

up the parts as they come off the conveyor belt. Rubber-faced rollers exerting a light pressure on the pieces force them forward to a second drive roll. The pick-up roll exerts relatively little pressure on the parts and imparts only a small amount of the forward motion. If too much pressure were exerted, the pieces might not enter because the conveyor belt exerts only a small longitudinal propelling force. A second drive roll provides most of the force necessary to propel the train of parts through the coil-quench fixture.

The feeder section can be adjusted either vertically or horizontally to accommodate shafts of different lengths and diameters. Relative to the longitudinal adjustment, the feeder is so adjusted on the V-ways that the pieces are always free of the second drive roll before entering the head stock of the scanner proper, where rotation is imparted.

Upon entering the head stock, the pieces are given rotary motion prior to passing through the coil-quench assembly. This can be seen in *Fig.* 3 which shows the entire scanner unit including head stock, coil-quench assembly and tail stock. The two lower rollers on both the head stock and the tail stock are fixed on ball bearing centers and are driven through a gear train. The upper rollers, in both stocks, are idlers. These upper rollers, however, are adjustable and, as in the case of the driver rollers in the feeder assembly, are spring-loaded to compensate for oversize and eccentric shafts. The complete spring assembly is adjustable vertically.

The upper idlers can be skewed so that, in addition to rotary motion, the piece can be given forward motion. This same set-up is duplicated in the tail stock, where the rollers take hold of the part before it is completely free of the head stock. In this way, it

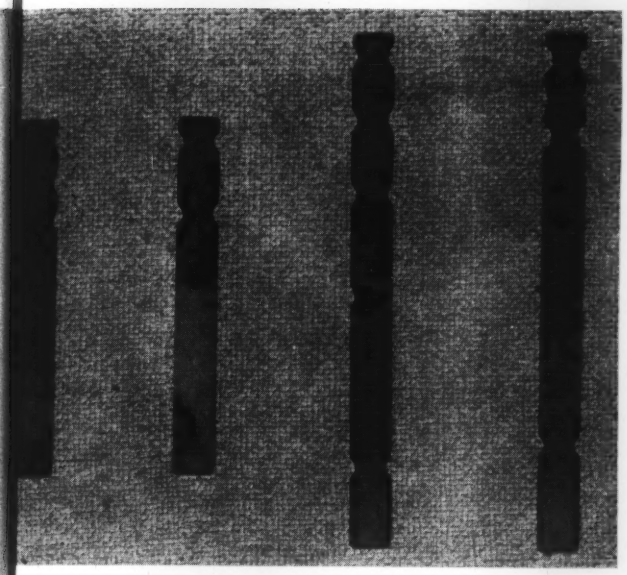


Fig. 4—Typical hardness patterns, shown by sectioned and etched automobile shifter rails. Selection of correct timer control circuit gives any hardness pattern

is always centered in the inductor coil and quench applicator, insuring a uniform concentric case. For shafts with splines and deep grooves, the rollers can be replaced with collets to insure smooth longitudinal and rotary movement through the coil-quench assembly.

Head and tail stock assemblies are movable on the V-ways to accommodate different length shafts and different coil-quench assemblies. The movable feature also facilitates set-up and maintenance.

Inductor coil and spray-quench applicator assembly is mounted between the head and tail stocks. Heavy, low-inductance copper leads carry the radio-frequency current to the coil and act as the supporting members of the assembly. These leads connect with the secondary of a current transformer through a slotted output terminal mounted in the bedplate of the machine. The primary current transformer leads are connected directly to the radio-frequency generator.

To harden rails selectively, *Fig.* 4, a selective timer mechanism is included in the machine. As the rails leave the tail stock of the scanner, they operate a micro-switch mounted over the out-feed guide trough. This switch triggers the timers which, in turn, key the generator so that power is automatically applied as selected shaft areas pass through the inductor coil. These timing circuits also control the solenoid-operated selectors in the magazine feed mechanism to allow another shaft to feed into the conveyor belt at the proper time.

At the tail stock end of the scanner is a "zero speed" switch. This switch is a roller-operated tachometer generator which, when it is at zero speed, actuates a relay which shuts off all power instantly. The roller can be seen at the right in *Fig.* 3. This is a most important protective device. If the train of pieces should stop for any reason, and the radio-frequency power not be cut off immediately, the piece in the coil would melt and jam the machine.

With the control switch mechanism at the discharge end of the scanner, it is necessary that a continuous line of parts be run through before the generator power and timing circuits function properly. To achieve this, several brass dummy pieces can be used to fill the loader magazine partially before starting a production run. After passing through the scanner, they can be placed in a special dummy chute located below the magazine for steel pieces. Part of this chute is counter-weighted so that at the end of a production run these dummies automatically pass through the scanner and insure proper driving force to clear the machine of steel pieces.

To simplify the set-up operation for changing from one shaft to another, the proper timing circuits can be automatically selected by making electrical connections between any given magazine and the timers. Each magazine has an electrical plug attached to a chain which, when inserted in the control circuit socket of the control timers, makes connection only with the desired timers to key the generator at the proper time for the particular timing circuit used.

Giant Trucks Speed Steel Handling

TWO of the world's largest ram trucks, capable of lifting 60,000 lb of steel coils, have been built by the Automatic Transportation Co. for the Great Lakes Steel Co. Overall height of the trucks is 155 inches, and at the broadest point they are 119 inches wide. Speed when loaded is 2.75 mph, and when empty, 3 mph. The "Giant" is equipped with two electrically differentiated dual drive units, each having two 24-inch tires of 36-inch diameter, a total of 96 inches of rubber across the truck.

Two disk brakes are mounted on the ends of the motor shafts to work through the gear train for more effective braking. Brakes are spring applied and hydraulically released with the hydraulic steering system controlled through a four-way valve.

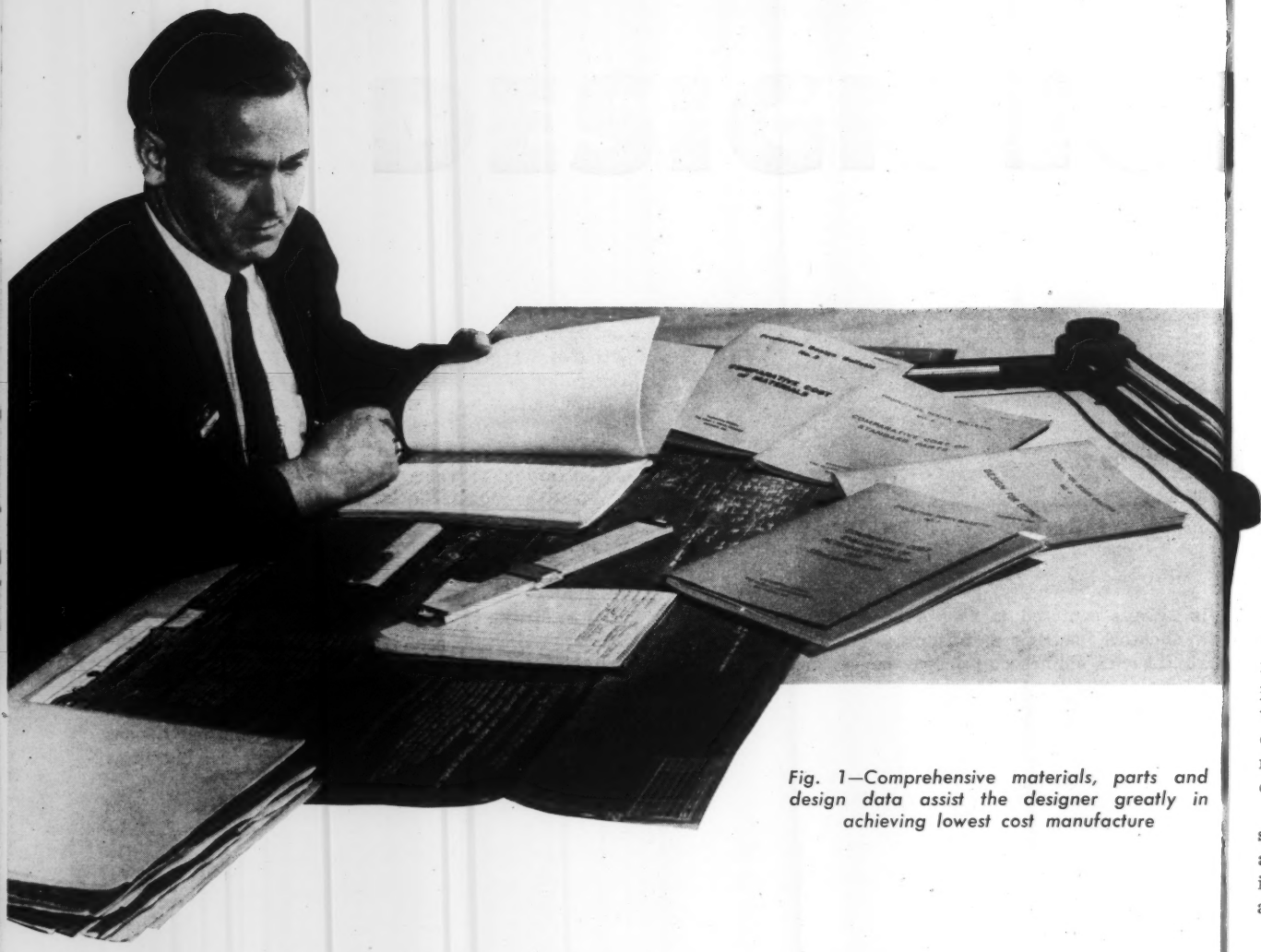


Fig. 1—Comprehensive materials, parts and design data assist the designer greatly in achieving lowest cost manufacture

TO MAINTAIN design and manufacturing efficiency in today's competitive market, it is essential that the complete design progression cycle be recognized and fulfilled. The general pattern of design evolution usually encountered in developing a product is: (1) Analysis of functional design aspects to determine working relationships; (2) investigation of material requirements, specifications and costs; (3) survey of design and manufacturing producibility in relation to contract quantity; and (4) incorporation of tooling requirements, types and procedures.

Examination of the functional aspects of design indicates a requirement of inherent qualities that a designer possesses from his natural aptitude for engineering and his educational background. However, the aspects of design for manufacturing producibility are not inherent but are phases of design that need constant investigation, development and application. And these are the basic fundamentals for achieving low-cost products.

The Glenn L. Martin Company has been actively engaged for a number of years in a development program to find ways and means to effectively step up the efficiency of designing to acquire economical production. One of the first steps in this effort was the development of design bulletin information, a basic function of the Design Cost Control Group. Shown in *Fig.* 1 is a fanfare of the first three basic types of bulletin information on "Comparative Cost of Materials", "Comparative Cost of Standard Parts", and "Design for Economy."

COMPARATIVE COST OF MATERIALS: This bulletin contains cost data on all metallic materials used in the fabrication of parts. It covers a range of material types such as bar stock, sheet stock, tubing, extrusions, castings, and forgings, in all commonly used alloys of aluminum, steel and magnesium.

All material cost data are shown in useful graphical form to facilitate obtaining comparative costs of each kind of material under each basic type. For bar stock, sheet metal, aircraft tubing and aluminum extrusions, *Figs.* 2 and 3, a premise of 100 to 300 lbs. is established as the normal purchased quantity of

Evaluation of materials, standard parts, design and assembly costs in the light of quantity for most economical manufacturing

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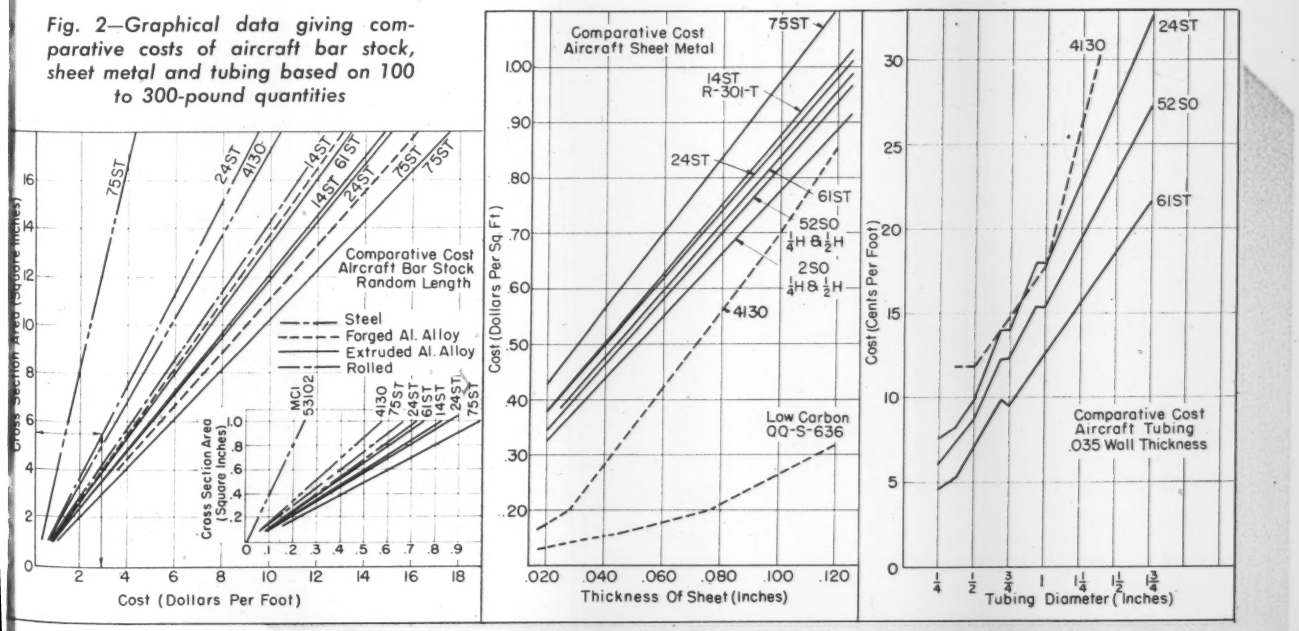
material to show the comparative cost relationships. The forging and casting cost data, *Figs.* 4 and 5, required considerable investigation and development. Classification premise as shown evolved into the definitions of simple, average and complicated classes to provide the engineer with an efficient means of obtaining accurate cost data for his selections. Also, a quantity premise of 100 pieces, established as a maximum purchase quantity, set the basis for the development of the curve plots.

Extruded material is widely used in aircraft design because of its excellent structural applications and ease of fabrication. To aid the engineer in arriving at a proper decision, a typical set of cost data as shown in *Fig.* 6 is utilized. This series of curves was prepared to show the cost of structural angles in terms of production methods and number of feet per production run. To prepare these curves, the following steps were taken: First, from previous purchase records, a list of frequently used angles was drawn up. By listing angle dimensions it is found that, by grouping, they fall into various diameter circumscribed circles. Second, by selecting the average size within the circle diameter, the basis for determining comparative production costs can be developed. As indicated in *Fig.* 6, crossover points show when certain methods of extrusion procedures pay for themselves, depending on the quantity required.

COMPARATIVE COSTS OF STANDARD PARTS: The sec-

Fig. 2—Graphical data giving comparative costs of aircraft bar stock, sheet metal and tubing based on 100 to 300-pound quantities



ALUMINUM ALLOY EXTRUSIONS

COMPARATIVE COSTS (BASED ON 100 TO 299 POUNDS)
Cost/Pound (Dollars) For Lengths of 1 to 29.999 Feet ①

DIE COST NOT INCLUDED

FACTOR ②	3S ③ AS EXT.	14S ③ AS EXT.	14S W	14S T	24S ③ AS EXT.	24S T	53-61S ③ AS EXT.	53S T5	53S 61S W	53S 61S T	63S ③ AS EXT.	63S T5	75S ③ AS EXT.	75S T
1 thru 4	0.61	0.695	0.715	0.725	0.715	0.76	0.62	0.63	0.665	0.675	0.61	0.62	0.83	0.89
5 thru 7	0.613	0.71	0.73	0.74	0.73	0.775	0.625	0.635	0.67	0.68	0.613	0.623	0.855	0.915
8 thru 10	0.616	0.73	0.75	0.76	0.75	0.795	0.63	0.64	0.675	0.685	0.616	0.626	0.88	0.95
11 thru 13	0.619	0.75	0.77	0.78	0.77	0.82	0.635	0.645	0.68	0.69	0.619	0.629	0.91	0.99
14 thru 16	0.622	0.77	0.79	0.80	0.79	0.85	0.640	0.65	0.69	0.70	0.622	0.632	0.94	1.035
17 thru 19	0.626	0.79	0.81		0.81	0.88	0.645	0.655	0.70	0.71	0.626	0.636	0.98	1.085
20 thru 22	0.630	0.81	0.82		0.835	0.91	0.655	0.665	0.71	0.72	0.63	0.64	1.03	1.14
23 thru 25	0.634	0.83	0.85		0.865	0.945	0.67	0.68	0.72	0.735	0.634	0.644	1.08	1.20
26 thru 28	0.639	0.855	0.885		0.895	0.985	0.685	0.695	0.745	0.755	0.639	0.65	1.14	1.27
29 thru 31	0.651	0.89	0.925		0.93	1.03	0.705	0.72	0.775	0.79	0.651	0.662	1.205	1.35
32 thru 34	0.673	0.925	0.97		0.97	1.075	0.73	0.745	0.81	0.825	0.673	0.685	1.285	1.44
35 thru 37	0.708	0.985	1.015		1.035	1.15	0.77	0.79	0.86	0.875	0.708	0.721	1.37	1.55
38 thru 40	0.743	1.045	1.08		1.10	1.225	0.81	0.835	0.91	0.925	0.743	0.757	1.49	1.66
41 thru 43	0.791	1.11	1.145		1.17	1.30	0.86	0.885	0.965	0.985	0.791	0.807	1.62	1.81
44 thru 46	0.839	1.19	1.225		1.24	1.39	0.91	0.935	1.02	1.045	0.839	0.857	1.76	1.97
47 thru 49	0.887	1.28	1.30		1.35	1.48	0.96	0.985	1.075	1.105	0.887	0.907	1.91	2.14

① For cost of lengths from 30 to 40 ft., all alloys, add $0.02 per lb.
For cost of lengths from 40 to 90 ft., all alloys, add $0.04 per lb.

② Factor = $\frac{\text{Cross section perimeter}}{\text{Cross section area X 1.2}}$ NOTE:- Express section factor to nearest whole number, decimals of .5 and over being taken to the next higher whole number.

③ For cost of "O" Temper, add $0.01 per lb. to "As extruded" cost.

④ If a new die is required, the cost must be amortized by dividing it by the total pounds of extrusion and adding this increment to the cost per lb. listed above to arrive at a true cost/lb. figure. Costs for hollow or semi-closed shapes, and stepped extrusions are subject to special inquiry.

DIE COST ④

*CIRCUMSCRIBED CIRCLE DIAMETER (INCHES)	SOLID SHAPES	ROD & BAR
Under 3	$100.00	$ 75.00
3 to (not incl.) 4	110.00	80.00
4 to (not incl.) 5	130.00	100.00
5 to (not incl.) 6	190.00	150.00
6 to (not incl.) 7	225.00	175.00
7 to (not incl.) 8	325.00	250.00
8 to (not incl.) 9	425.00	325.00
9 to (not incl.)10	500.00	375.00

* The circumscribed circle diameter is the diameter of the smallest circle that will completely enclose the cross section of the shape.

Fig. 3—Table giving cost data for extrusions

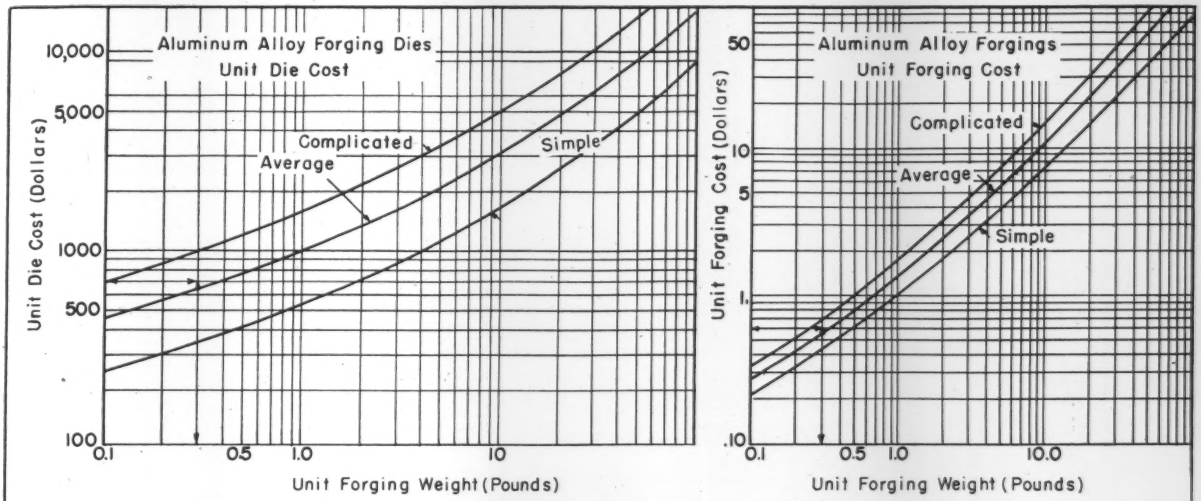
ond bulletin lists costs of standard parts such as rivets, bolts, screws, and fasteners. Almost every manufacturing company uses standard parts in the fabrication of a product and, of these, there are many sizes, types, etc. Usually the installed cost of a standard part, rather than initial cost, determines whether it can be used or not. To facilitate presenting this information for the engineer's use, it is organized into two main divisions: (1) Installed cost of standard parts used in the general applications of structural designs, and (2) installed cost of standard parts used in general applications of nonstructural designs. All standard parts are listed by their specific commercial or AN standard name, number, size, length, grip, weight, and installed cost. In the total cost are the cost of the part, the labor cost for handling, drilling, reaming, countersinking, dimpling, riveting, deburring, tightening, checking by operator, and all other operations required to complete the installations.

DESIGN FOR ECONOMY: This bulletin is comprised of over 125 classified fitting summary sheets as typically shown in *Fig.* 8. This type of presentation shows the designer the aspects of manufacturing and tooling costs on the type of material selection for the quantity of fittings under consideration, As a series of historical references of fitting designs, these summary sheets are of great value to the designer. Using these references for comparing similar shape, size and weight to a newly designed fitting will enable the designer to make a design decision with a fair degree of accuracy.

Quantity is a Key Factor

The procedure followed in obtaining the data itemized in *Fig.* 8 is typical of the co-ordinated efforts of the designer and Design Cost Control engineer in reaching the correct decision. Four methods of fabricating this fitting are analyzed to determine the most economical method of manufacture for the type of material selected: (1) machined 24ST bar stock, (2) machined 14ST forging, (3) welded 4130 steel, and (4) machined aluminum-alloy casting. Premise is set for a quantity of 100 parts, with shop-lot releases set at 25 parts. This premise, as in any cost investigation, is of vital importance since the *quantity* of parts to be made has a *direct* bearing on the method of manufacturing.

The first step in the analysis is to develop material



Forgings Classification

Simple: (1) Impression in one half of the die.
(2) Parting planes are straight.

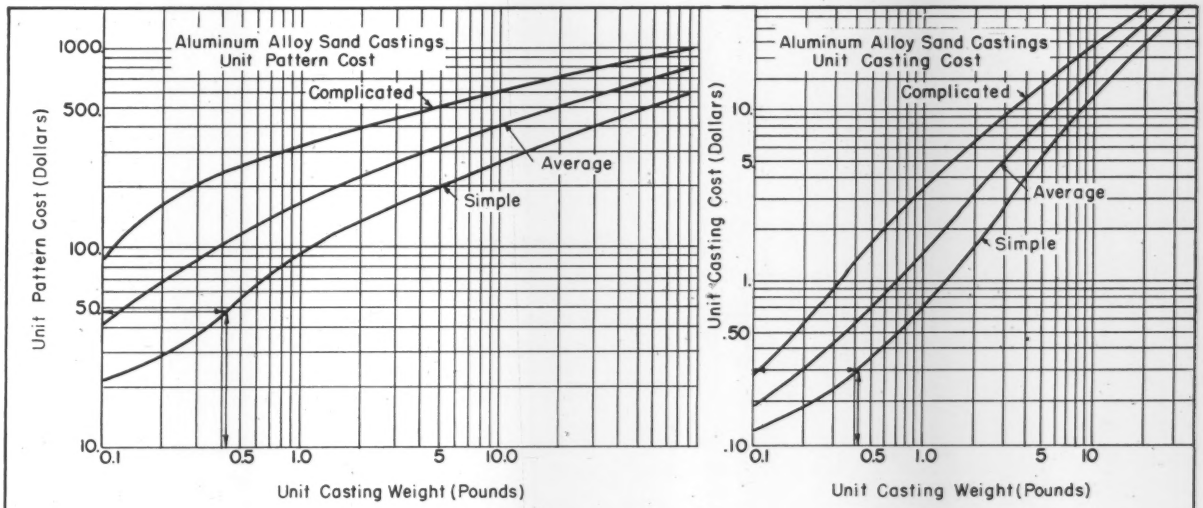
Average: (1) Impression in both halves with straightforward die sinking.
(2) Parting planes are straight.

Complicated: (1) Impression in both halves of the die plus complicated or curved die faces of wrapped parting plane.
(2) Deep impression to fill.
(3) Multiple blocking dies necessary.

General Notes: Die costs remain constant for any quantity of pieces.
The unit price is subject to change for low quantities.
Die setup charges approximately 10 percent of die cost.

Fig. 4—Above—Charts showing unit forging costs and unit forging die costs based on 100 pieces. Classification basis of curves is explained in note

Fig. 5—Below—Sand casting and pattern unit costs plotted against weight for quantities of 100 pieces. Note shows classification basis



Casting Classification

Simple: (1) Impression in one half of the mold.
(2) No coring

Average: (1) Impression in one half of the mold plus simple coring.
(2) Impression in both halves but no coring.

Complicated: (1) Impression in both halves of the mold plus intricate coring

Note: In quantities of less than 100 pieces the pattern equipment is usually "loose wood" of pine or mahogany material and runs approximately 40 to 60 percent of permanent metal equipment.

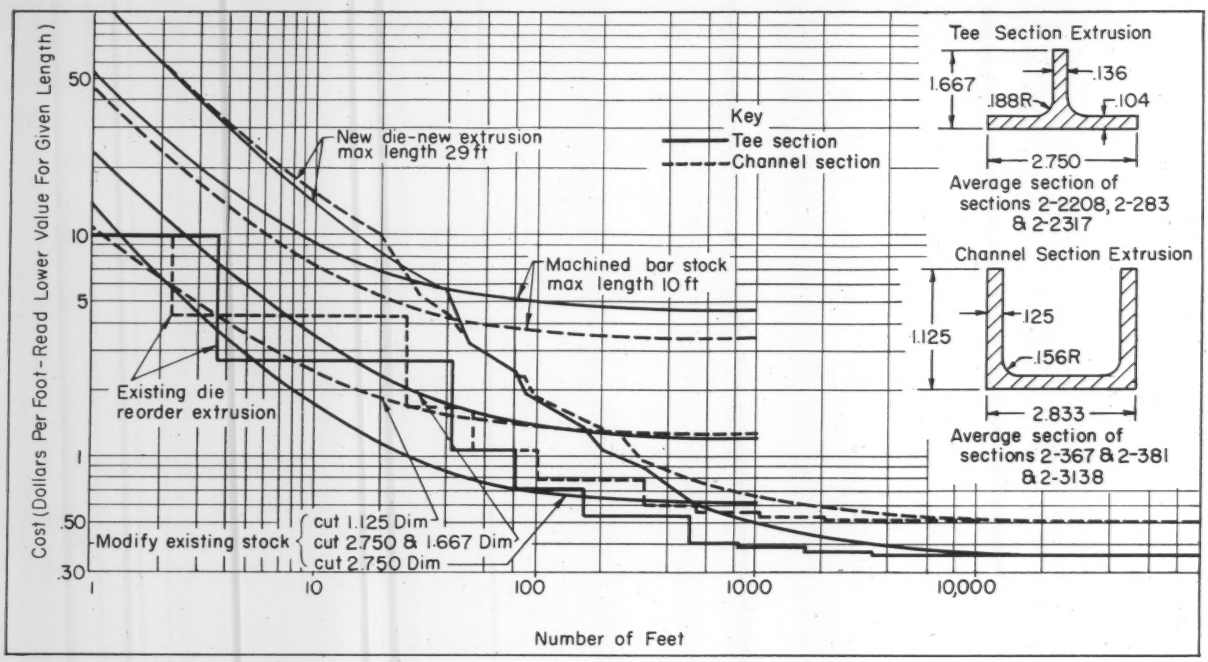
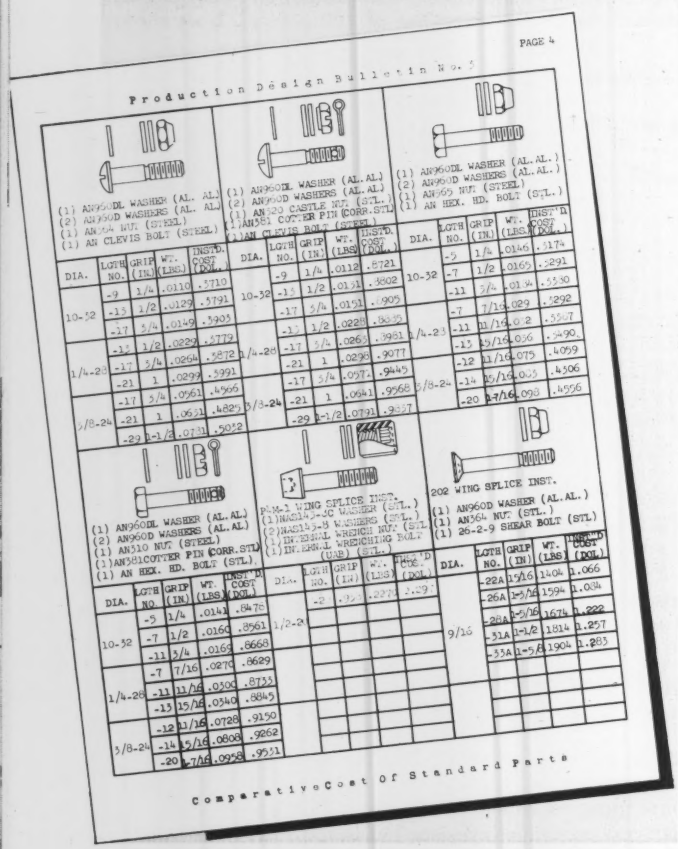


Fig. 6—Above—Comparative costs for 24ST machined bar stock tees or channels and extrusions within 3 to 4-inch circumscribed circle. Costs based on contract quantities

Fig. 7—Below—Cost comparisons of typical standard parts. Installed cost is used rather than merely the piece cost



costs. The unit material and forging die or pattern costs for each application is readily obtained from the charts in the bulletin "Comparative Costs of Material" as typically shown in *Figs.* 2, 3, 4, and 5.

The next step in the analysis is to develop the manufacturing cost by means of an elemental breakdown of setup and labor time in machining or fabricating each design. To develop these elemental breakdowns requires an experienced man who knows all types of machines used in the shop as well as the techniques used in producing a part from these machines. The time for each operation is evaluated from standard data books compiled by the Cost Control Group. Adding the elemental time values for both setup and run time establishes the total operation time sequence. Normally this total in minutes is changed to hour values for conversion to dollar costs.

Dollar Conversion Factor Evolved

The evaluation of the dollar cost conversion factor requires the incorporation of cost accounting data for rework and rejection allowances, direct labor immediate supervision adjustment factor, and average departmental efficiency percentage. Then by applying the average direct labor rate and burden percentage, the complete dollar-hour conversion factor is evolved. This figure, when multiplied by the amortized setup and total run times in hours, will give the final dollar cost of fabrication.

After the material and fabrication cost of each fitting is determined, the next step is the consideration of the tooling cost to complete the total cost evaluation. The fixed tools necessary to complete the fabrication are composed of a prick punch template for locating the attachment holes and a welding fixture. All four fitting designs require the prick punch tem-

plate with only the welded design requiring the additional welding fixture. The type of tools and manufacturing hours to build them are co-ordinated with the tool planning engineers.

One more cost item called Specification Test, *Fig.* 8, must be applied to the manufacturing cost of the Class "A" casting, since this casting is classified as a structural fitting. The testing cost is amortized over the quantities of purchased castings. This testing cost is composed of structural engineering and laboratory costs.

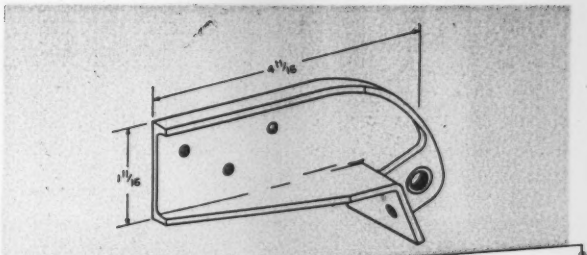
Combining the material, fabrication and testing cost gives total unit manufacturing cost for each fitting: Bar stock $10.99; (2) forging $10.39; (3) weldment $4.19 and (4) casting $7.92. The weldment proved to be the most economical for the quantity of parts under investigation. A critical review of this typical cost breakdown, *Fig.* 8, for each method of fabrication enables the engineer to determine why certain designs are more costly than others so that he can take the necessary steps to modify the design to insure economical production.

As emphasized in the early part of the fitting analysis, quantity of parts to be made has a direct bearing on the method of manufacturing and the type of material selected. The importance of the statement is graphically portrayed in *Fig.* 9 which shows the quantity of parts against manufacturing costs. With up to 4 parts, the machined bar stock fitting is the recommended design, with the welded fitting the most economical design from 4 to 520 parts. Beyond this number of parts the casting becomes the best method of manufacturing.

Efficient Assembly a Design Problem

The efficient assembly of this fitting to the airframe structure is another controlling factor where the designer can express his ingenuity and initiate an economical method of attachment. In this particular attachment problem, the fitting is subjected to large structural loads to support the operating action of the landing gear. Checking with the Stress Group, it is found necessary to use 4 AN-3-5 bolts, 3 washers and elastic stop nuts, also 3 AN-4-7 bolts with the same number of washers and stop nuts. Studying the listed bolt cost, *Fig.* 7, the selection indicates that the four AN-3-5 bolts cost $0.32 each or a total of $1.28 and the three AN-4-7 bolts at $0.33 each, totaling $0.99. Thus, the total installation cost for attaching the fitting to the frame structure will be $2.27. The cost of hinge bolt AN-6-20 with 3 washers, castellated nut and cotter pin costs $0.95 each to attach the door to the hinge fitting. The designer can further expand his search of installation cost, if the Stress Group will permit, by suggesting a rivet assembly and comparing this cost with the bolted assembly.

Design histories presented in this manner help to establish standardization of design and serve as a stepping stone for the development of future designs. It also provides a basis for comparing the design features and cost evaluations of these problems to achieve a scientific approach to sound engineering practices.



STUDY NO.:
DATE:

COST CONTROL GROUP
PRODUCTION DESIGN SECTION—ENGINEERING DIVISION

COMPARATIVE COST SUMMARY SHEET MODEL: P4M-1

TITLE: FITTING ASSY. - MAIN GEAR OLEO DOOR AFT. HINGE

DRAWING NO.: 20-21-0715

ANALYSIS: 24ST Machined Bar Stock - VS - 24ST Machined Forging - VS Welded Fitting H.T. to 125,000 P.S.I. - VS - Al. Alloy Casting Class "A"

PREMISE: Total of 100 parts. Production lots of 25 parts per lot. Tool & labor hours were coordinated with representatives from Tool Engineering & Detail Manufacturing.

SUMMARY:

ITEM	BAR STOCK		FORGING		WELDED		CASTING CLASS "A"	
	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST	TOTAL COST	UNIT COST
MAT'L		1.56		.58		.19		.50
PURCHASED EQUIPMENT								
FIXED TOOLS DETAIL	19.51	.19	19.51	.19	50.17	.50	19.51	.19
FIXED TOOLS ASSEMBLY								
DETAIL SET-UP	14.67	.59	8.65	.35	18.56	.75	8.65	.35
DETAIL LABOR		8.35		2.18		2.76		2.18
ASSEMBLY SET-UP								
ASSEMBLY LABOR								
DIE			706.75	7.09			48.00	.48
PATTERN								
ENGINEERING COST							442.40	4.42
SPEC. TEST								7.92
TOTAL UNIT COST		10.99		10.39		4.19		
UNIT WEIGHT	.229 LBS.		.521 LBS.		.457 LBS.		.417 LBS.	

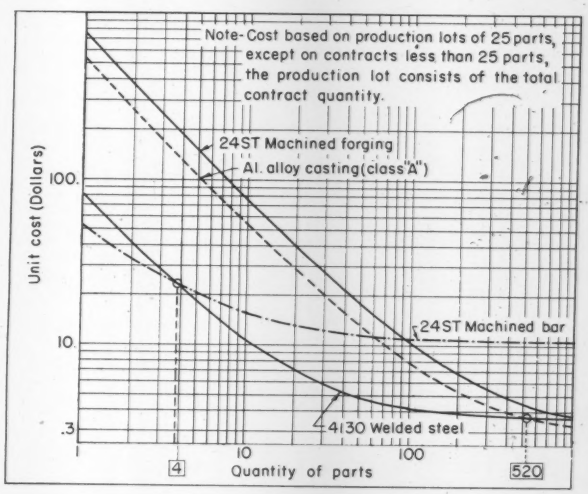
REMARKS: As indicated by the above figures, the welded fitting proves to be the most economical to manufacture. However, to realize this savings a weight penalty of .208 LBS/PART would have to be accepted.

ACCEPTANCE OF COST DESIGN DECISION: DATE:

PREPARED: DATE: APPROVED: DATE:

Fig. 8—Above—Breakdown of costs for producing the hinge fitting by four different manufacturing methods

Fig. 9 — Below — Fitting costs by four different methods plotted against quantity shows importance of this factor

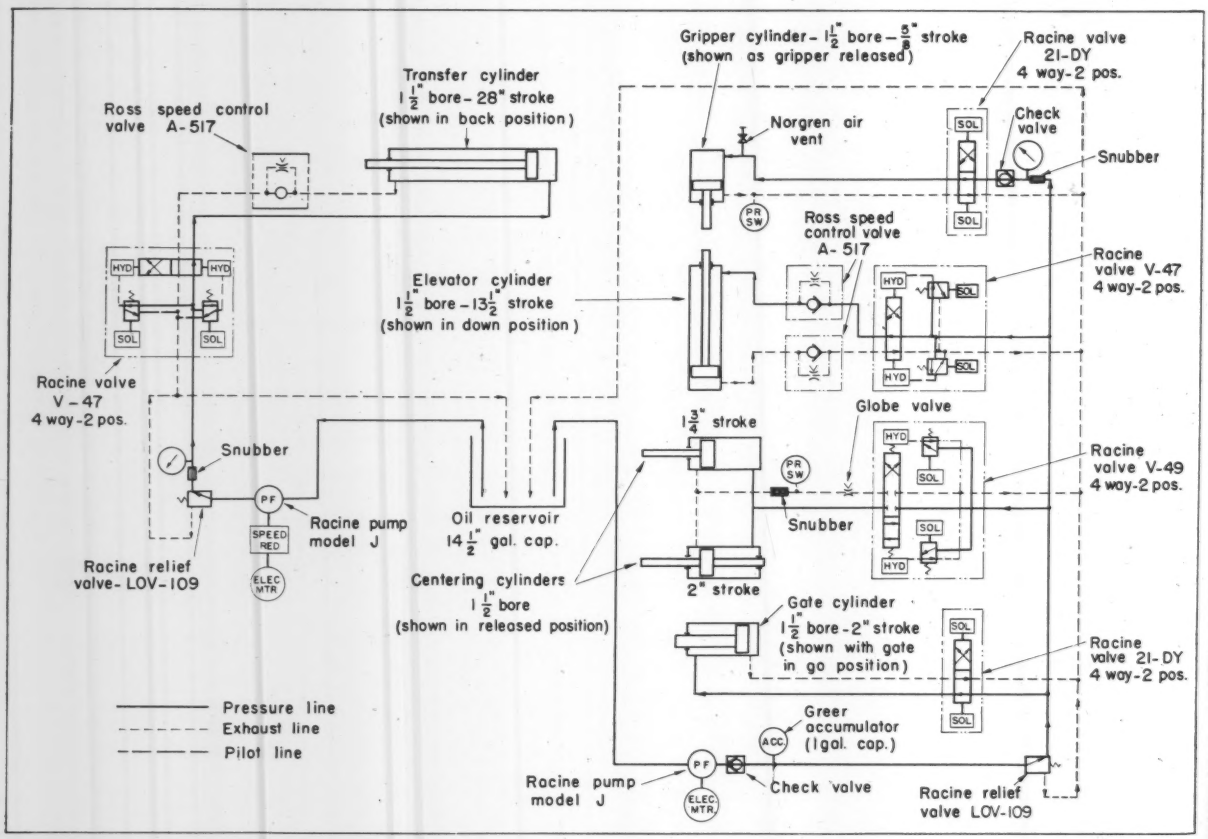


High-Speed Case Unpacker

SEQUENCE of operations in the automatic case-unpacking machine, far right, is electrically controlled to prevent start of any phase of the unloading cycle until the preceding operation has been completed. Production model of the unpacker, made by Edward Ermold Co., removes a variety of standard containers from cartons, deep or shallow cases and trays, all automatically. Solenoid valves control operation of the hydraulic cylinders that actuate major components of the unpacker, as shown in the schematic diagram of the hydraulic system, below. Symbols used in this drawing are those contained in the JIC hydraulic standards (MACHINE DESIGN, December 1948).

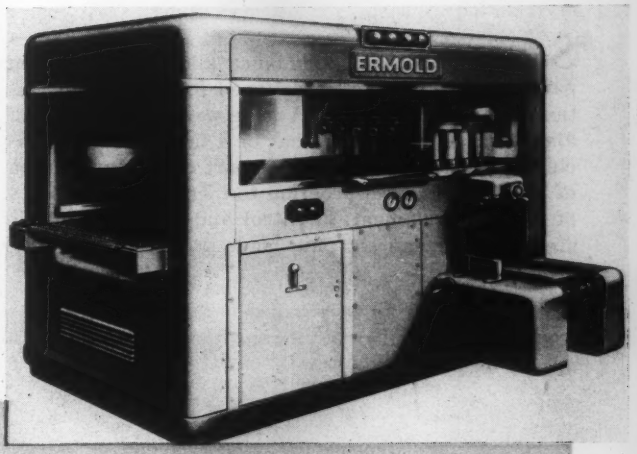
Cases arriving on a feed conveyor, either singly or in an unbroken line, are admitted one at a time through a gate onto the power-driven case conveyor. Another case is not admitted until the preceding one has been unpacked and discharged from the machine. The full case is then gripped by longitudinal and lateral centering arms (drawing bottom right) which square the case in the machine and hold it while the conveyor slips beneath it. A vertically-reciprocating elevator, positioned below the level of the case conveyor, lifts the case free of the centering arms. In the extreme top elevator position, the necks of the bottles or other containers enter the open grip fingers. Hydraulic action closes the fingers and the elevator descends with the case, leaving the containers suspended from the gripper carriage. Strippers pull any cases free that might be stuck to the containers.

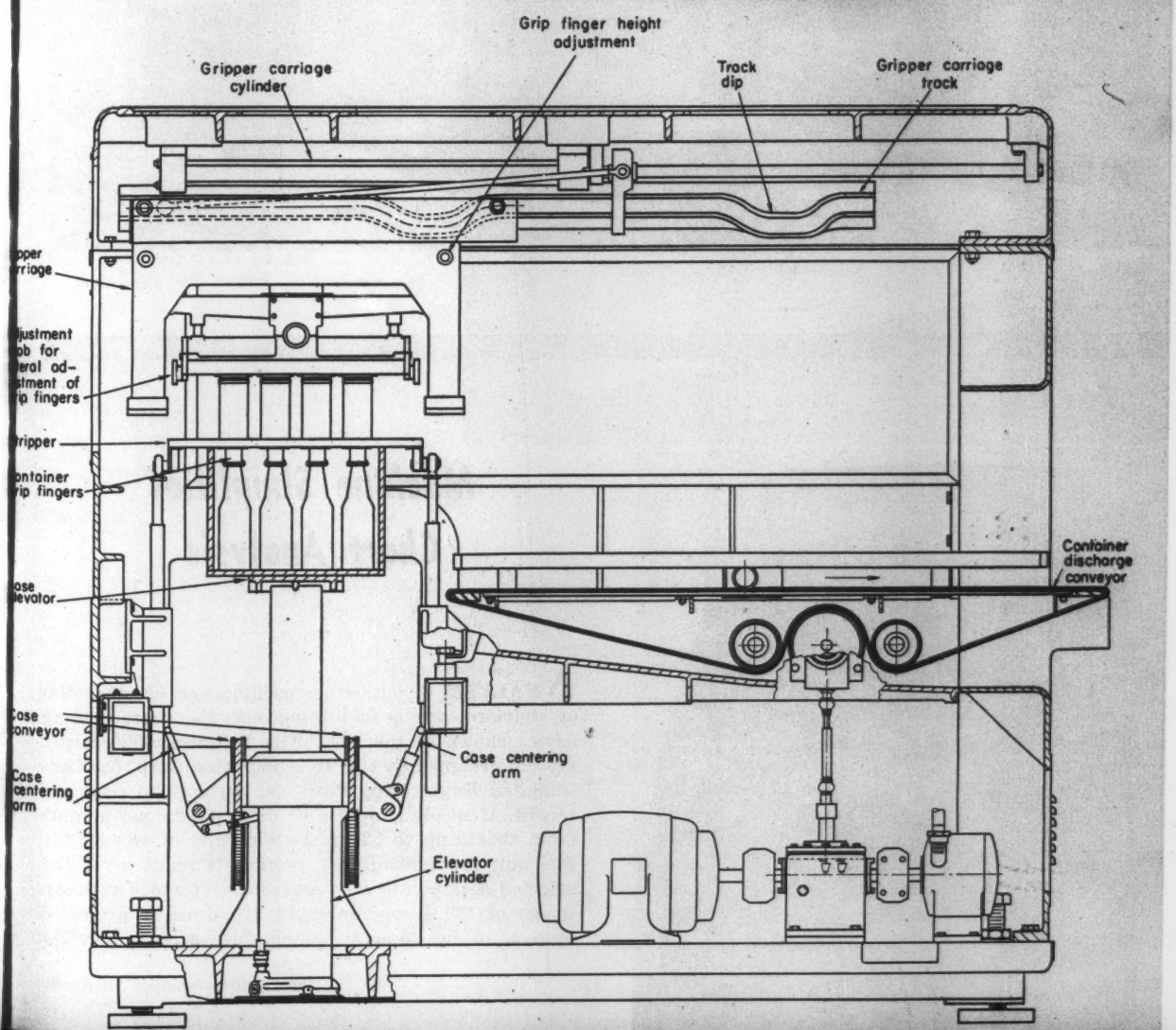
The gripper carriage travels on a track at right angles to the motion of the input conveyor, drawing


Gripper cylinder – 1½" bore – ⅜" stroke (shown as gripper released)
Racine valve 21-DY 4 way-2 pos.
Transfer cylinder 1½" bore – 28" stroke (shown in back position)
Ross speed control valve A-517
Norgren air vent
Check valve
Snubber
Ross speed control valve A-517
Racine valve V-47 4 way-2 pos.
Elevator cylinder 1½" bore – 13½" stroke (shown in down position)
Racine valve V-47 4 way-2 pos.
Globe valve
1¾" stroke
Racine valve V-49 4 way-2 pos.
Snubber
Oil reservoir 14½" gal. cap.
Racine pump model J
Racine relief valve-LOV-109
Centering cylinders 1½" bore (shown in released position)
2" stroke
Gate cylinder 1½" bore-2" stroke (shown with gate in go position)
Racine valve 21-DY 4 way-2 pos.
Greer accumulator (1 gal. cap.)
Racine pump model J
Check valve
Racine relief valve LOV-109
Pressure line
Exhaust line
Pilot line

DESIGN

below. The track is so positioned that bottoms of the containers are slightly above the level of the wire-mesh discharge conveyor. Near the end of the carriage track, the carriage dips to bring the containers down on the conveyor. Grip fingers are then opened and bottles move out of the machine on the discharge end. Speeds vary from 4 to 15 cycles per minute.





Scraper Designed For Air Transport

SUBSTITUTION of aluminum alloys for steel in parts of a motor grader designed for air transportation, below, have resulted in a weight reduction of almost three tons under that of a normal-weight machine. This redesigned unit is part of a Navy Bureau of Yards and Docks program to increase the mobility of service equipment. Control and operation of the steering gear and scraping blade are hydraulic. The grader is driven at any one of six forward speeds between 1.9 and 15 mph and in reverse at 1.9 or 6.5 mph by a 6-cylinder, 4-cycle, 80 brake- horsepower diesel engine.

Provisions have been made for adding weight to the grader, when on the job, if more than its 17,000 lb bare weight is desired. Manufacturer: Austin-Western Co., Aurora, Ill.

Machine Simplifies Chart Analysis

ANALYSIS of multitrace oscillograph or other film or chart records is facilitated with the chart reading device shown at the left. The Telereader-Telecorder system designed by the Telecomputing Corp. for Consolidated Engineering Corp. can be used to read and record, if desired, up to 40 data points per minute from charts up to 12 inches wide and of any length. The unit provides digital co-ordinate readings of any selected data points with respect to reference axes. An image of the record enlarged 2½ times is projected onto a 15 by 30-inch ground-glass screen, together

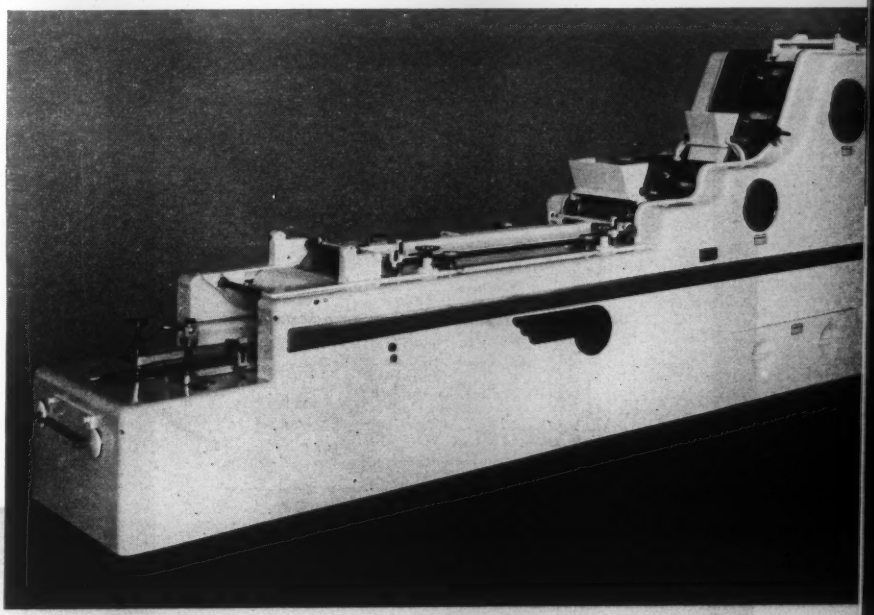
with an image of a system of two movable crosswires. The location of these crosswires is adjusted manually by the operator to the point on the chart being recorded. When the operator presses a button, wire positions are recorded on binary electronic counters to 0.001-inch. The numbers on these counters are then punched into IBM cards for subsequent sorting or tabulation.

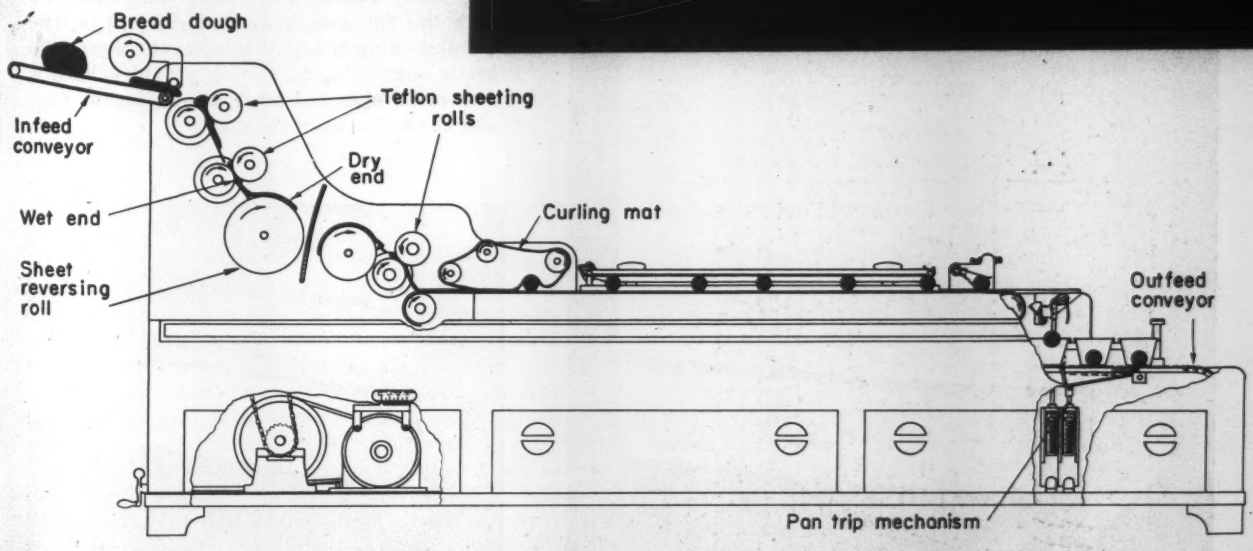
Two optical systems are provided: a reflection system for opaque paper charts and a transmission system for transparent film charts. Chart speed is controlled by a foot pedal between ½ to 1200 inches per minute forward and 12 to 120 ft per minute reverse. Readings may be taken with the chart moving continuously at a slow rate, or the chart may be stopped for each reading. Generally, readings can be duplicated, from sharp records, within plus or minus 0.005-inch. Zero backlash in the gearing actuating the crosswires permits approaching the data points from either direction with equal accuracy.

Bread Molder Improved By Welded Design

AUTOMATIC machine, right, for molding bread dough into loaves and depositing them in pans for baking, employs a welded frame and cover construction. Smooth, flat surfaces are easily cleaned and eliminate places for dough or flour to collect—important prerequisites in any baking operation. The Moulder-Panner, made by Read Machinery Div. Works of The Standard Stoker Co. Inc., is unusual in that it reverses the sheet of dough during the sheeting operation (drawing, below), improving the interior grain of the loaf and keeping the bread fresh longer.

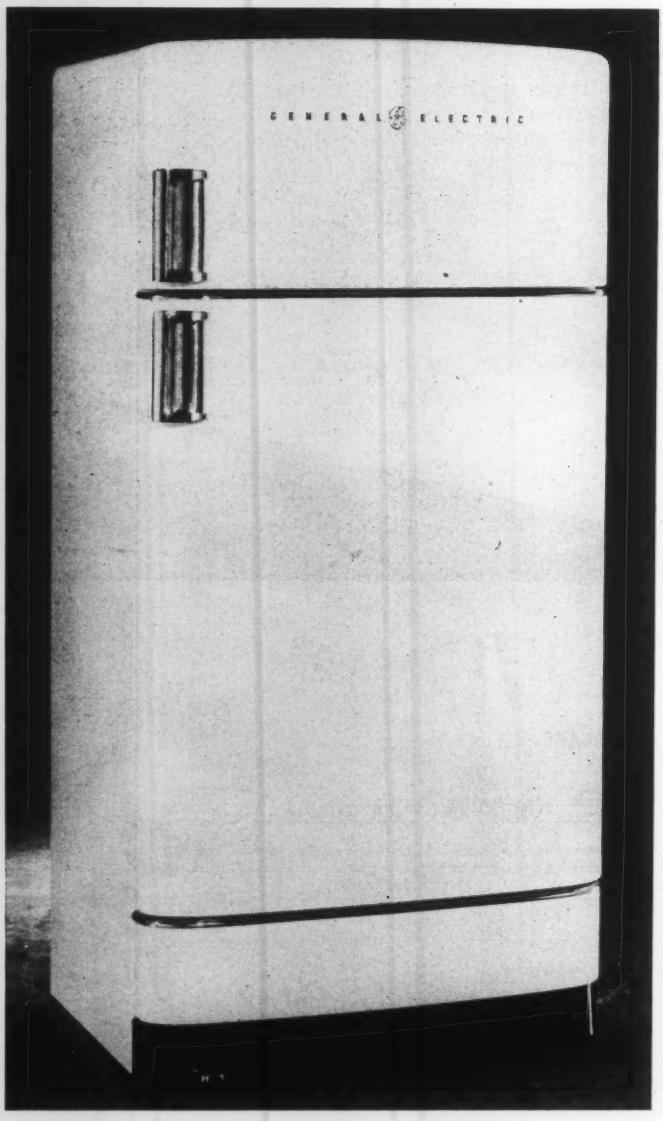
Two sets of Teflon rolls presheet





the dough as it comes off the infeed conveyor. With rolls made of this material, tight-fitting scrapers which heat the rolls and raise the temperature of the dough pieces are not necessary. Because dough does not stick to Teflon, little or no dusting flour is required. The sheeted dough piece is reversed by two flip-over rolls which turn the dough end-for-end in mid-air. After another sheeting operation, the dough is carried under a power-driven curling mat which curls the "wet" end inside to give the desired texture. After passing under a pressure board which molds the loaf to the required shape, the dough is dropped into pans, controlled by an adjustable trip mechanism, and released to the outfeed conveyor. Speed of the machine, which handles ¾ to 3-lb loaves from 6 to 16 inches long, is adjustable to suit any dough condition.

Structural frame of the Moulder-Panner is fabricated from 3-inch channel and 1½-inch angles. The 13-ft by 31-inch frame requires about 400 welds made with approximately 50 lb of ⅛-inch and 5/32-inch Lincoln Fleetweld No. 7 electrode. The formed cover sheets are largely welded 16-gage cold-rolled sheet.



Magnetism Seals Refrigerator Door

MAGNETIC attraction between the steel cabinet and Alnico permanent magnets in the door of the two-compartment General Electric refrigerator, left, do away with the conventional door latch. Closing of the door is accomplished quietly and a tight, uniform seal is obtained by the one-inch long magnets contained in the seal gasket. Both the upper freezing compartment and the lower fresh-food section are equipped with the magnets, only a slight pull on the handles being required to open the door. The lower compartment is also provided with a foot-pedal operated lever, shown below, that can be used to open the door.

Locating Axis of Rotation of a Moving Link

By Fred H. Posser
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Fig. 1—Above—The axis about which an aircraft landing gear should rotate in retracting into the wing is critically affected by stability and structural considerations. This axis is simply and accurately determined by the method outlined in the article

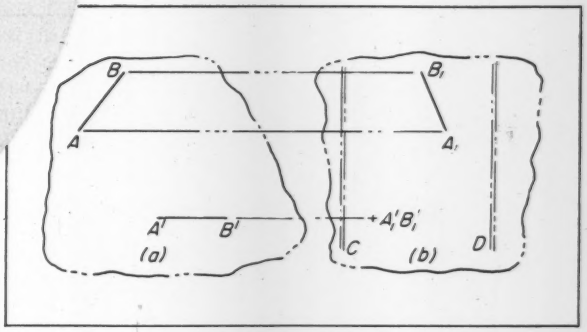


Fig. 2—Problem is to determine the axis about which line AB rotates in moving to position A'B'. Right-hand view shows projections of the two positions on a plane at right angles to the left-hand view

IN MACHINE and mechanism design the engineer often runs into the problem of determining the point about which a member will rotate. Usually the solution, whether by layout or analysis, is not difficult except when it is necessary to locate a skewed axis accurately or to locate its supporting bearings in the main frames of the unit.

By the use of the method explained in this article any skewed axis can be located easily and accurately. The type of problem usually encountered is one in which two positions of the member are known and it is necessary to determine the axis about which the member rotates.

A good illustration of the use of the method is the determination of the axis of rotation of aircraft retractable landing gears, *Fig* 1. In an aircraft landing gear of the full cantilevered type, retracting inward or outward into the wing, the extended and retracted positions of the wheel are extremely critical. The extended position is determined by ground stability factors and must be properly located with respect to the center of gravity of the airplane. The retracted position is determined by the structure surrounding the area into which the wheel must retract. The distance from the spindle axis to the wheel will in all probability be from four to six times the distance between the bearings of the spindle. Any

small error in laying out the axis will vary the location of the wheel either extended or retracted by an appreciable amount.

This problem is therefore one in which end positions of the motion are located within close limits. The required axis is one that will give the two positions of the wheel.

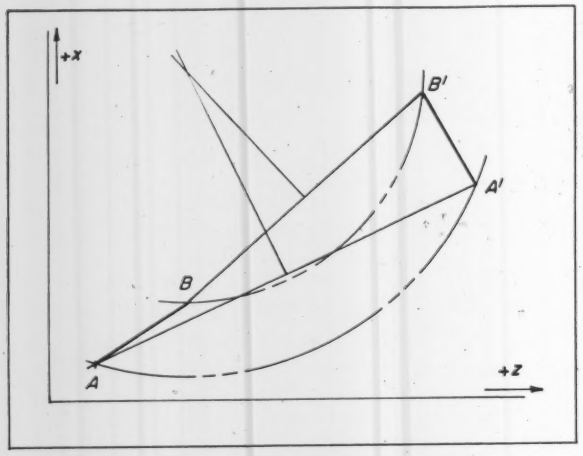
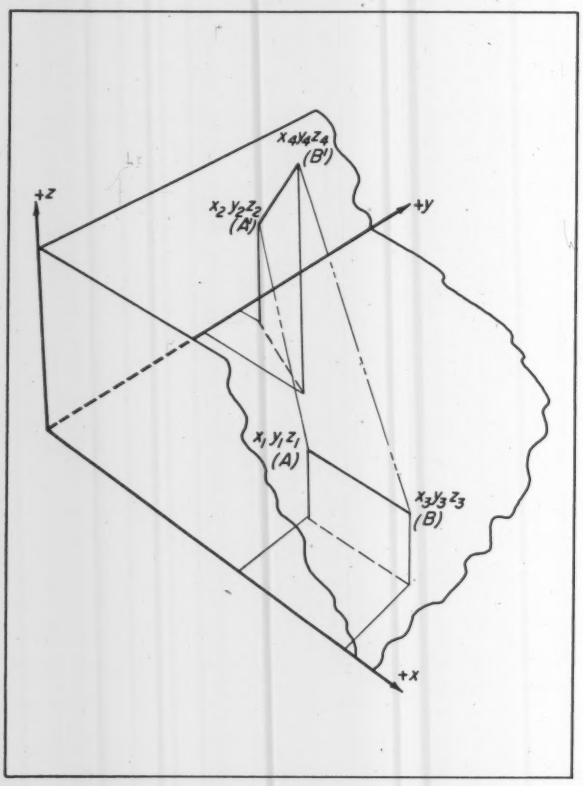


Fig. 3—Above—As line AB rotates to position A'B' in the plane of the paper, the center of rotation lies at the intersection of the perpendicular bisectors of lines AA' and BB'

Fig. 4—Below—When initial and final positions of a line in the rotating member are in different planes, problem must be set up with reference to a three-axis system



Lines AB and $A'B'$, *Fig.* 1*a*, represent two positions of the same member. *Fig.* 2*b* is a side view of *Fig.* 2*a*; A_1B_1 and $A_1'B_1'$ are projections of AB and $A'B'$.

It is required that the member be attached to a rotating shaft in such a manner as to produce the motion shown. It is also necessary to mount the shaft bearings in two frames, *C* and *D* in *Fig.* 2*b*.

Using the foregoing information it is possible, using various layout techniques, to determine the centerline of the bearings. It is also possible to calculate this axis by mathematical means. The layout method is usually not accurate enough and most mathematical methods involve more manipulation than is necessary. The following method requires only simple, straightforward mathematics and is extremely accurate.

It is a familiar fact that if a body rotates so that all points on it describe circles in any one view, the axis will be perpendicular to the view. The axis will lie on the perpendicular bisector of a line joining any two positions of one point on the body. If any two positions of two points are used the axis will lie at the intersection of the two bisectors.

In *Fig.* 3, AB and $A'B'$ represent two positions of a line in a member which always remains in the plane of the paper. The center about which the member rotated in going from AB to $A'B'$ must be at the intersection of the perpendicular bisectors of lines AA' and BB'.

If the entire problem were set up with reference to a co-ordinate system in one plane as shown in *Fig.* 3, the equations of the perpendicular bisectors, and therefore the co-ordinates of the axis, could be determined mathematically.

In the more general case line AB or line $A'B'$, or both, are not necessarily in the plane of the paper, hence AA' and BB' may not lie in the plane of the paper. The previously discussed theory is still valid but must be expanded slightly. The bisectors of AA' and BB' will no longer be seen as lines but will appear as planes. The intersection of these two planes will be a line, which is the axis about which the body rotates. The entire problem is again set up with reference to a co-ordinate system, this time made up of three axes, as shown in *Fig.* 4.

If $x_1y_1z_1$ are the co-ordinates of A and $x_2y_2z_2$ are the co-ordinates of A', the equation of the perpendicular bisector of line AA' is:

$$(x_2 - x_1)\ x + (y_2 - y_1)\ y + (z_2 - z_1)\ z - \tfrac{1}{2}\ (x_2^2 - x_1^2 + y_2^2 - y_1^2 + z_2^2 - z_1^2) = 0 \quad \ldots\ldots\ldots (1)$$

In a like manner, if $x_3y_3z_3$ are the co-ordinates of B and $x_4y_4z_4$ are the co-ordinates B', the equation of the perpendicular bisector of BB' is:

$$(x_4 - x_3)\ x + (y_4 - y_3)\ y + (z_4 - z_3)\ z - \tfrac{1}{2}\ (x_4^2 - x_3^2 + y_4^2 - y_3^2 + z_4^2 - z_3^2) = 0 \quad \ldots\ldots\ldots (2)$$

If Equations 1 and 2 are solved simultaneously and x set equal to zero, the resulting equation will be the projection of the required axis on the yz plane. If y is set equal to zero the resulting equation will be the projection of the axis on the xz plane, and

(*Concluded on Page* 196)

Why Machine Parts Fail

Part 3—Modes of Failure

By Charles Lipson
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Photo, courtesy Chicago Pneumatic Tool Co.

MACHINE parts may fracture for many different reasons but the actual mechanism of failure usually follows either of two basic modes:

1. Ductile or shear type of failure, caused by excessive shear stresses and resulting in slip along crystallographic planes
2. Brittle or normal type of failure, caused by excessive normal stresses and resulting in separation of the crystal along planes normal to the maximum tensile stress.

Successful analysis of a given fracture will depend upon recognition of the mode of failure. This in turn depends upon a thorough understanding of the two modes and factors affecting them.

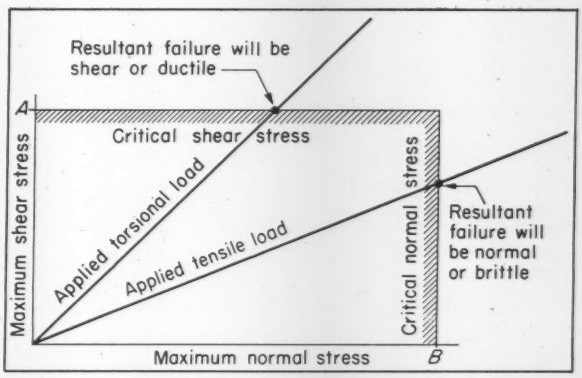
WHICH FAILURE TO EXPECT: The particular mode of failure that will prevail in any given application is the result of the comparative balance between two sets of opposing forces. On the one hand are shear stresses opposed by resistance of the metal to slip. On the other are normal stresses opposed by cohesive or "tear-apart" resistance of the metal. Which of the two systems of opposing forces is overbalanced first determines the mode of failure.

Some materials can fail in a ductile manner under one set of conditions and in a brittle manner under another set. One of the most common materials exhibiting this phenomenon is mild steel, which exhibits ductile characteristics if pulled in a simple tensile test but evidences a brittle fracture when subjected to fatigue loading or when severely notched.

Another example is that of tool steels, which under pure torsion fail in a ductile manner but under simple tension fail in a brittle manner. This case is illustrated in *Fig.* 24 where maximum shear stress is plotted as the ordinate and maximum normal stress as the abscissa. Critical shear stress—the shear strength of the tool steel—is shown as *A*. Critical normal stress—the tensile strength of the tool steel—is shown as *B*. The rectangular boundary based upon these two values represents the strength.

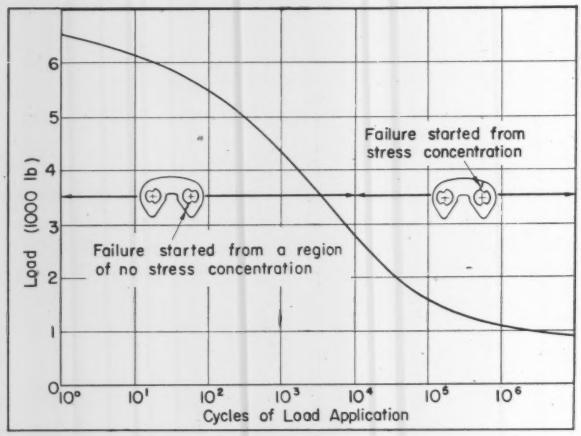
When a machine member is subjected to a pure torsional load the maximum shear stress equals the maximum tensile stress. The line defining this loading condition will, therefore, take a slope of one or

Fig. 24—Effect of loading on type of failure in tool steels is represented by this diagram



a 45-degree direction, as shown. This line will intersect the critical shear stress boundary, giving rise to a shear or ductile type of failure. On the other hand, if the same steel is subjected to simple tension, shear stress will equal one-half the tensile stress and the slope of the line will be one-half, as shown. This line will intersect the critical normal stress boundary and normal or brittle failure will occur.

Fig. 25—Effect of number of load applications on type of failure is shown by these data for 2½-in. pitch silent chain links which were not heat treated. Up to approximately 10,000 cycles, failure was ductile; above 10,000 cycles, brittle. Data supplied by courtesy of Warner Automotive Parts Div., Borg-Warner Corp.



The particular rectangle shown in *Fig.* 24 refers to tool steels. Similar rectangles can be drawn for other materials. For each rectangle the mode of failure will be determined by a variety of external conditions.

All external conditions can be analyzed for their effects on the mode of failure. The most important factors in determining which type of failure will take place are:

1. **External Factors Affecting Mode of Failure**
   a. Manner of loading—static or fluctuating
   b. Range of imposed stresses
   c. Number of load applications
   d. Velocity of loading
   e. Direction of stress—uniaxial or polyaxial
   f. Temperature during load application
   g. Geometry of the part—presence of stress concentrations
2. **Internal Factors Affecting Mode of Failure**
   a. Material
   b. Surface treatment of material.

MANNER OF LOADING: A machine member that is ductile under static loading will fail in a brittle fashion when the load is fluctuating. This is effectively illustrated by data given in *Fig.* 25 for silent chain links tested from one cycle to ten million cycles. It will be noted that for loads applied from one to approximately 10,000 times failure took place across the small section of the link. This is consistent with the view that when load is applied only a few times the condition is similar to static loading for which failure is determined by the nominal stress (section size) rather than by stress concentration.

Photo, courtesy Harry Ferguson Inc.

Fig. 26—This rear axle shaft failed from fatigue after running 1,439,000 miles at four times the normal loading

Photo, courtesy Briggs Manufacturing Co.

Fig. 27—Bending caused this fatigue failure. Note the clear indication of the course taken by the crack

As the number of load applications increased, failure took place across the larger section of the link, starting from the region of stress concentration. In this case the deleterious effect of the notch was more than sufficient to offset the advantage of the larger section. At the notch the state of stress was no longer uniaxial but triaxial, a condition which led to brittle failure.

Transition from ductile to brittle failure in this test appeared to occur in the neighborhood of 10,000 cycles. Approximately the same value was obtained from another series of tests on the same type of links which, unlike samples in the first test, were heat treated and preloaded.

The character of brittle failure with a complete absence of ductility is indicated by fractured samples illustrated in *Figs.* 26 and 27. Characteristic features of brittle (fatigue) fractures, such as the stress waves in *Fig.* 27, will be interpreted in subsequent articles.

TYPE OF STRESS: Repeated stresses do not always result in fatigue failure, because fatigue is affected by the range of stress and the number of applications. A variable stress, other than complete stress reversal, is caused by an alternating stress superimposed upon a steady stress. If the ratio of alternating stress to steady stress is relatively low, it may still be possible to have a yielding type of failure, or if the number of repetitions is low, the failure may also fall within the plastic classification. Often there is no distinct line of demarcation since results are affected by the material and the stress history (previous amount of cold working, for example). *Figs.* 28 and 29 indicate the effect of stress range on mode of failure.[1]

SPEED OF LOADING: Velocity of loading is another factor which determines whether a given material will fail in a ductile or a brittle manner. A material that evidences ductile failure under simple tension at ordinary testing speeds may show brittle failure when loaded at high velocity. It should be noted that in notched impact tests, such as the Charpy, it is the presence of the notch and not the speed of loading that causes brittle fracture. To produce a brittle fracture in a ductile material requires speeds of 50 to 60 feet per second.

According to laboratory tests, internal resistance to slip increases directly with the velocity of deformation, but resistance to normal separation is not similarly affected. Although zinc, for example, will bend with marked ductility under slow loading, it will fracture in a brittle fashion under sudden loading. *Figs.* 30 and 31 illustrate the effect of speed of loading of energy absorption (ductility).[2]

DIRECTION OF STRESS: Preceding examples have shown how a ductile material can fail in a brittle fashion. An example of how a brittle material can be made to show plasticity is given by Von Karman. He demonstrated that marble, when subjected to hydrostatic (polyaxial) pressures, would deform plastically instead of crumbling as it would under uniaxial compression.

The opposite effect is found in the simple tensile

[1] References are tabulated at end of article.

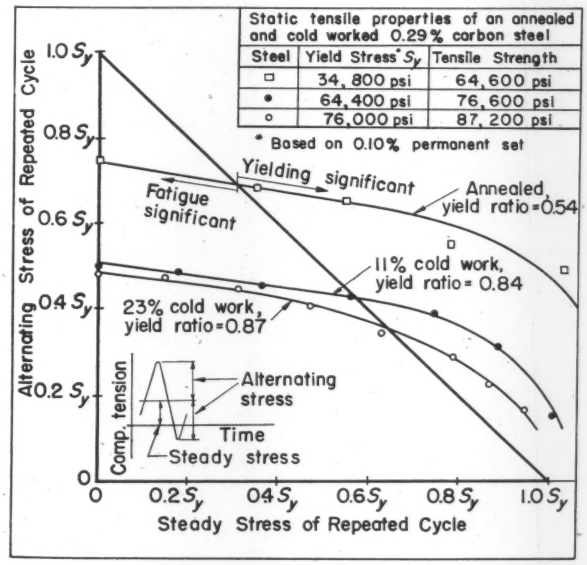

Static tensile properties of an annealed and cold worked 0.29% carbon steel
Steel | Yield Stress* S_y | Tensile Strength
□ 34,800 psi 64,600 psi
• 64,400 psi 76,600 psi
○ 76,000 psi 87,200 psi
* Based on 0.10% permanent set
Alternating Stress of Repeated Cycle
1.0 S_y
0.8 S_y
0.6 S_y
0.4 S_y
0.2 S_y
0
Yielding significant
Fatigue significant
Annealed, yield ratio=0.54
11% cold work, yield ratio=0.84
23% cold work, yield ratio=0.87
Comp. tension
Alternating stress
Time
Steady stress
0 0.2 S_y 0.4 S_y 0.6 S_y 0.8 S_y 1.0 S_y
Steady Stress of Repeated Cycle

Fig. 28—Effect of stress range on mode of failure for low-strength steel is shown by this data reported by Smith. Note range where fatigue failures are significant

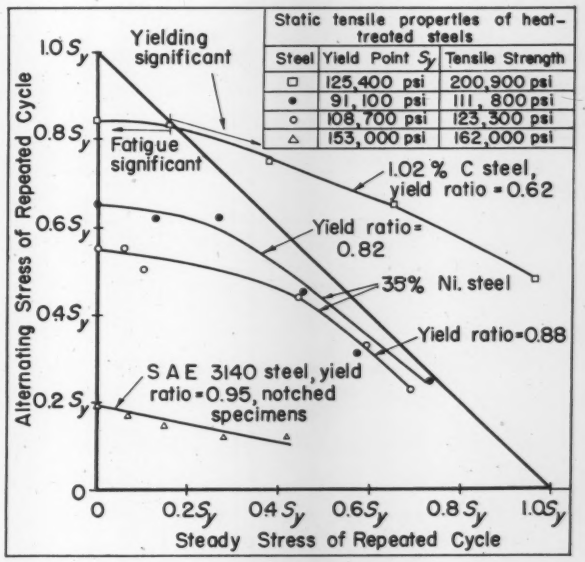

Static tensile properties of heat-treated steels
Steel | Yield Point S_y | Tensile Strength
□ 125,400 psi 200,900 psi
• 91,100 psi 111,800 psi
○ 108,700 psi 123,300 psi
△ 153,000 psi 162,000 psi
Alternating Stress of Repeated Cycle
1.0 S_y
0.8 S_y
0.6 S_y
0.4 S_y
0.2 S_y
0
Yielding significant
Fatigue significant
1.02% C steel, yield ratio = 0.62
Yield ratio = 0.82
35% Ni. steel
Yield ratio=0.88
SAE 3140 steel, yield ratio=0.95, notched specimens
0 0.2 S_y 0.4 S_y 0.6 S_y 0.8 S_y 1.0 S_y
Steady Stress of Repeated Cycle

Fig. 29—Paralleling Fig. 28, this graph by Smith shows effect of stress range on mode of failure for high-strength steels. Again yield tensile ratio affects fatigue range

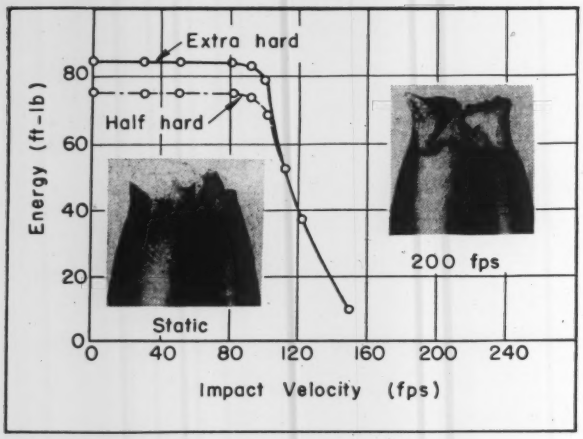


Fig. 30—Above—This study by Mann shows effect of impact loading. Note sharper necking (less ductility) of impact specimen compared to static specimen

Fig. 31—Below—Adapted from work by Mann, this graph shows relationship of impact velocity and energy absorption. Note the decreasing ductility with increasing velocity of load application

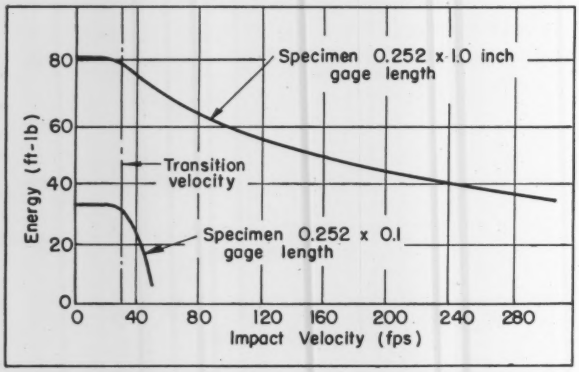
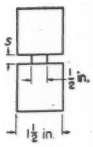


Fig. 32—Above—For this fracture, illustrated by Gensamer, the tensile test was stopped, before complete rupture. Copper specimen was sectioned to show failure originating as a brittle fracture in the center prior to shear failure of outer fibers

Table 1—Effect of Notch Severity on Ductility

Notched Specimen	Dimension S (in.)	Ultimate Strength Original Area (psi)	Ultimate Strength Reduced Area (psi)	Reduction of Area (%)
		Carbon Steel[1]		
	3/32	163,000	176,000	7.5
	1/16	164,000	177,000	7.5
	1/8	143,000	158,000	9.5
	0	102,000	227,000	55
		Nickel Chrome Steel[2]		
	3/32	193,000	237,000	18.5
	1/16	184,000	232,000	21
	1/8	154,000	199,000	23
	0	108,000	348,000	69

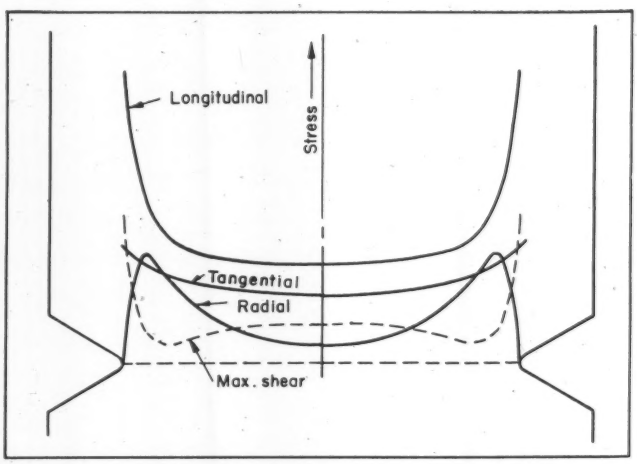
[1] Carbon steel: ultimate tensile, 102,000 psi; yield, 64,500 psi; proportional limit, 56,000 psi; elongation 26.5%; reduction of area, 55%.

[2] Nickel-chrome steel: ultimate tensile, 108,000 psi; yield, 85,000 psi; proportional limit, 80,000 psi; elongation, 27%; reduction of area, 69%.

test of mild steel. Upon examination, a fracture of this type will be found to be made up of a flat central area with a surrounding conical band, illustrative of central cohesive failure and external ductile failure. During the pulling process the tensile specimen necks down, whereby the stress changes from uniaxial to triaxial. The result is a cohesive fracture in the core, *Fig.* 32, which continues outward until completion of shear failure in the outer area.[3]

STRESS DISTRIBUTION: As some of the previous examples have shown, any discontinuity in material or in shape of the machine member that causes locally triaxial stresses may alter the type of fracture from plastic to brittle by increasing the normal stresses and reducing the shear stress. This may occur microscopically because of an inclusion and initiate a crack which will lead to brittle fracture under fatigue loading. It may happen on a macroscopic level resulting

Fig. 33—Below—Diagram shows variation of state of stress across a notched cylinder subjected to tension



from a sudden change in diameter of the piece, such as in a severely necked bar, and cause a brittle fracture. An example of the effect of notch severity on ductility is given in TABLE 1.[4] Any stress concentration factor such as a notch, sharp tool mark, or a change in section can affect the condition of stress and therefore the mode of failure. *Fig.* 33 is a graphical representation of the three dimensional stress system in a notched bar in tension.

TEMPERATURE CONDITIONS: Temperature of metal during loading appears to affect its resistance to slipping. In general, resistance to slip increases with low temperatures and decreases with high temperatures. It is clear, therefore, that a brittle fracture can be produced in a ductile material by lowering the temperature sufficiently so that cohesive forces are overcome under load before slip resistance breaks down. However, under elevated temperatures a plastic flow may take place at stresses considerably lower than the conventional yield point because resistance to internal slip is lowered. The effect of temperature, state of stress, etc., was recently investigated by Jones and Worley.[5]

PERIOD OF LOAD APPLICATION: The effect of time is essentially the same as the effect of temperature, in that the allowable stress is lowered if the time during which the load is applied is prolonged. *Fig.* 34 shows the effect of time of load application on the type of failure.[6]

TYPE OF MATERIAL: It is obvious that inherent structural properties of the material will always play an important role in determining the type of failure. Because a certain type of failure is frequently associated with a certain material, many materials have come to be classified ductile or brittle. However, external loading conditions may be such as to alter the expected mode of failure. As a general rule, any material will fail when either the resistance to slip or the cohesive force is overcome by external stresses. In one or the other type of resistance some materials have relative advantages implicit in their microstructure as indicated in the first article of this series. In terms of the balance analogy, then, a fracture must be analyzed for mode of failure by weighing the inherent material properties against the external load conditions.

SURFACE TREATMENT: Various surface treatments introduce residual compressive stress in the surface of the part. This compressive stress counteracts tensile stress resulting from the externally applied load, thus reducing the probability of a cohesive failure. In this category belong shot peening, carburizing, induction hardening, nitriding, etc. It will be noted, however, that surface treated parts can develop failures originating either at the surface or below the surface, depending on the type of applied loading and the severity of stress concentration.[7] Various types of conditions for notched and unnotched bars are diagrammed in *Fig.* 35.

In the next article of the series, actual identification and classification of fractures will be discussed.

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2. H. C. Mann—"High Velocity Tension-Impact Tests," *Proc. ASTM*, Vol. 36, 1936.
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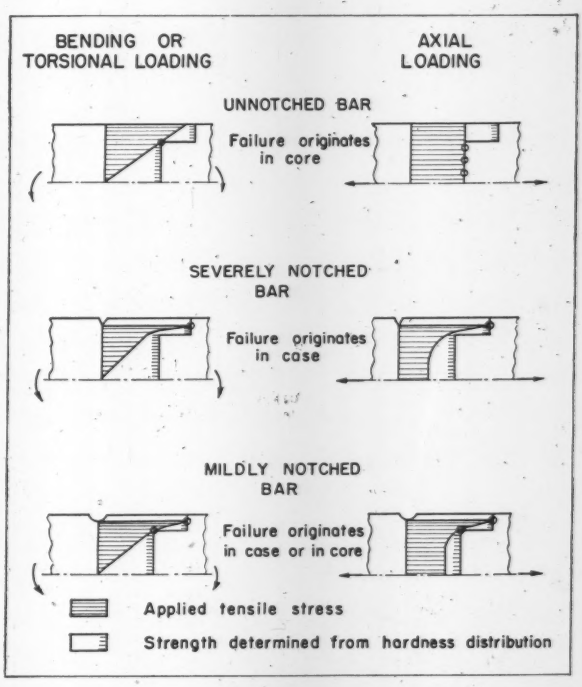
4. S. Timoshenko—*Strength of Materials*, Van Nostrand, New York, 1942, Page 422.
5. P. G. Jones and W. J. Worley—"An Experimental Study of the Influence of Various Factors on the Mode of Fracture of Metals," paper presented at Fifty-first anual meeting of ASTM, June, 1948.
6. C. W. Dollins—"Symposium on the Significance of the Tension Test," *Proc. ASTM*, Vol. 40, 1940.
7. C. Lipson, G. E. Noll and L. S. Clock—*Stress and Strength of Manufactured Parts*, McGraw-Hill, New York, 1950.

Fig. 34—Left—Lead sheath specimens, tested by Dollins, failed in different ways because of different rates of load application. Specimen (a), ductile failure, was loaded at low speed; specimen (b), brittle failure, was loaded at high speed

Fig. 35—Right—Failure of surface-treated parts is diagrammed for different types of loading. Although residual compressive stresses introduced at the surface counteract the tensile load, notch effects may cause initiation of fracture in the outer fibers



Drum Brakes

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VERSATILITY and prevalence of the shoe brake are attributable to its reliability, efficiency and effectiveness, and to the fact that its inherent simple design has made it possible to produce brakes of this type at low relative cost. This discussion covers three brake types identified here, for convenience, as Duo Servo, Twinplex and Non Servo types.

The Duo Servo brake, *Fig.* 1, is generally fabricated with a stamped steel brake support, or backing plate, to which are attached the brake shoes, hydraulic wheel cylinder, anchor pin, adjusting screw, and parking brake actuating linkage used only in rear brakes. Sufficient hydraulic pressure in the wheel cylinder forces both shoes into contact with the brake drum. Considering counter-clockwise drum rotation as shown, the shoe to the left is the primary and the other the secondary. The primary shoe leaves the anchor pin due to friction between the drum and the brake lining, and anchors on the adjusting screw, a floating link. This anchor load then becomes the applying force on the secondary shoe, which is forced into the drum with a load much greater than the wheel cylinder exerts on this secondary shoe. As a result, the drum carries both shoes around, with the secondary shoe anchoring on the anchor pin immediately above the wheel cylinder.

The secondary shoe is actuated with a greater force than the primary shoe and therefore would be expected to do a greater amount of work. With the same lining material on both shoes, the secondary lining would wear out before the primary lining. However, to equalize wear, dissimilar lining materials generally are used on the two shoes. The secondary lining usually has a lower coefficient of friction, better wearing properties, and is longer than the primary lining.

The second type, the Twinplex, incorporates a stamped steel backing plate, to which are attached two identical brake shoes, two wheel cylinders, two adjusting eccentrics, shoe hold-down springs and shoe return springs. *Fig.* 2*a* is a line drawing of this brake. When hydraulic pressure is built up in the wheel cylinders, both shoes pivot about their anchors and contact the brake drum surface simultaneously. The shoes anchor on machined surfaces of the wheel cylinder castings and return against the externally adjustable eccentrics.

It will be noted that the shoes are not anchored on pins, secured in the backing plate. Instead, the

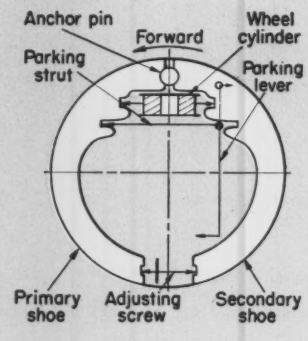


Fig. 1—Left—Sketch of Duo Servo rear brake showing single hydraulic wheel cylinder and anchor pin

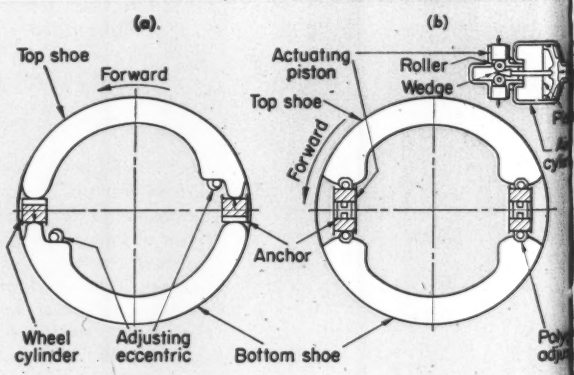


Fig. 2 — Right — Twinplex front brake illustrating two identical brake shoes with two actuating cylinders and eccentrics at (a), with wedge method of actuating brakes shown at (b)

BASIC DESIGNS and operating characteristics of drum and disk brakes, as abstracted from two papers presented at the SAE National Passenger Car, Body and Production Meeting in Detroit, March 14, 1950

rounded ends of the shoes anchor against flat surfaces, allowing the shoes to float and center themselves in the drum.

This "floating shoe" feature is to be found in the Twinplex and some constructions of the Non Servo brake. It has several advantages over the fixed anchor design: better centering of the shoe in the drum, particularly after the initial adjustment; slightly higher effectiveness; reduction of brake noise; and more economical construction.

It is obvious that work done by this brake is equally divided between the two shoes. However, the brake is more effective with forward drum rotation than with reverse drum rotation. With forward drum rotation, tangential force imposed on each brake shoe by the rotating drum adds to the actuating force from the wheel cylinders to create anchor loads greater than those obtained with reverse drum rotation. With reverse drum rotation, the tangential loads on the brake shoes work against the applying loads on the shoes to reduce anchor loads and resultant torque.

Equal effectiveness for forward and reverse braking, so far as front brakes are concerned, is not generally considered necessary in view of the relatively low vehicle speed and reduced front axle loads encountered in decelerating in reverse. Rear brakes, however, should be almost equally effective in forward and reverse braking.

The Twinplex brake makes use of the same length of lining segment, and the same lining material on each shoe to accomplish equal lining life.

Another construction of the Twinplex brake is shown in *Fig. 2b*. This is a mechanically-actuated, air-operated brake in use in the bus field. The brake differs essentially from the previously described brake in respect to the method of applying load to the brake shoes. Thrust from the air cylinders is imparted to the brake shoes by means of wedges forced between rollers in the actuating pistons, thereby providing the shoe thrust to engage the brake.

Adjustment of this brake is unique. It will be noted that polygon adjusters are interposed between the shoe ends and the actuating pistons. These adjusters are spool-like, with an extension through the dust shield to provide outside adjustment. Each pair of flat faces that contact the piston varies in distance from the centerline of the hub. Faces are progressively arranged so that when adjustment for lining wear is required, the adjusters are rotated as much as lining clearance allows. The shoe-return springs maintain sufficient pressure on the adjusters at all times to prevent rotation during brake actuation. Output of the brake can be varied by changing wedge angle, air cylinder diameter, and, of course, lining material and air pressure.

Non Servo Shoe Loads Unequal

The third brake type is the Non Servo, *Fig.* 3. In general, it consists of a stamped steel brake support, or backing plate, the two shoes, a single wheel cylinder, two adjusting eccentrics, shoe return and hold-down springs, and for the rear brakes, the parking brake strut and lever. When hydraulic pressure is built up in the wheel cylinder, both shoes are forced into the brake drum. Anchor loads are taken on the block located between the lower ends of the shoes. When equal force is applied to each shoe from the wheel cylinder, the forward shoe does more work than the reverse shoe with forward (counterclockwise) drum rotation. The reason for this is that the tangential force on the forward shoe is added to the

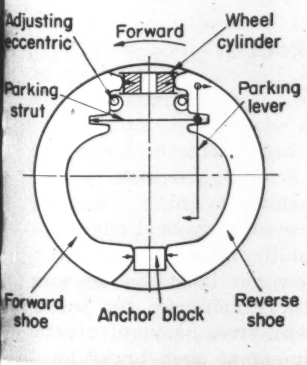
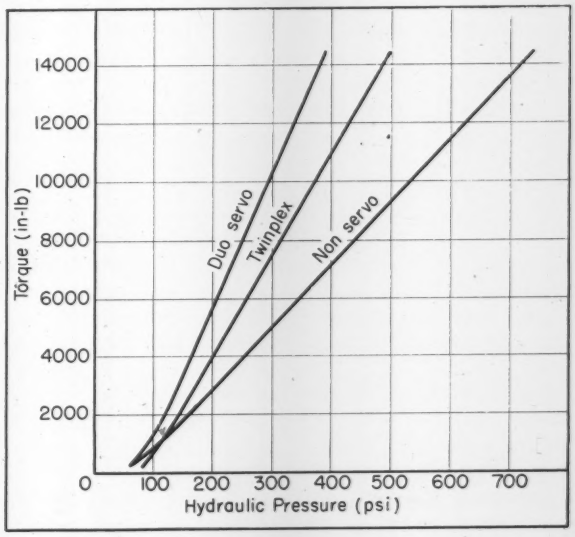


Fig. 3 — Left — Non Servo type rear brake showing single actuating cylinder and stationary anchor block

Fig. 4—Right—Curves showing relative effectiveness of three drum brake types



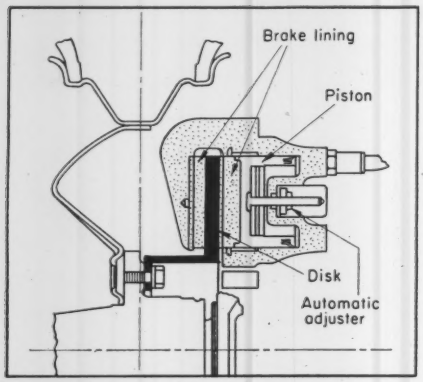
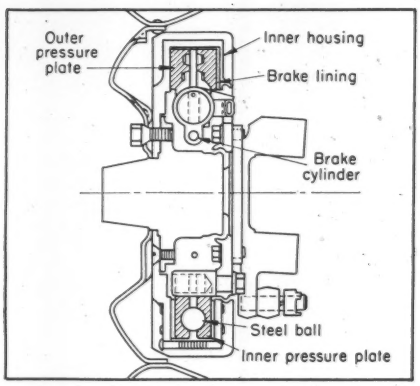


Fig. 5—Left—Cross section through Goodyear non-self-energizing disk brake shows small area of friction buttons

Fig. 6 — Right — Section through Auto Specialties Co. self-energizing Lambert disk brake



applying force to create a greater load on the anchor block than in the case of the reverse shoe, where the tangential force on the shoe is bucked by the applying force on the shoe. The applying force on the reverse shoe is always greater than the tangential force on this shoe; otherwise, the shoe would rotate away from the anchor.

When this brake is equipped with a straight-bore wheel cylinder and the shoes are lined with identical segments, lining wear will be more rapid on the forward shoe than on the reverse shoe, when the brake is used mostly in forward braking. Various methods can be used to reduce this difference in lining wear. The lining segment on the reverse shoe can be reduced in area by making it shorter in length, reduced in width, or both; dissimilar lining materials can be used on the two shoes; and a step-bore wheel cylinder can be used to provide greater thrust on the reverse shoe than on the forward shoe.

On the other hand, in order to get the most from the brake some vehicle manufacturers prefer to put the forward shoe to work by use of a step-bore wheel cylinder that exerts greater force on the forward shoe than on the reverse shoe. This, of course, aggravates unequal lining wear, but by juggling lining materials and lining areas, the breach can be reduced. In reverse braking the reverse shoe does the greater amount of work.

Generally, the effectiveness (ratio of brake output to input) of the Duo Servo is highest, the Twinplex is intermediate, and the Non Servo is lowest. Considering each brake on the basis of the combined work of its two shoes, the theoretical effectiveness is in the approximate ratio of 5 for the Duo Servo, 4 for the Twinplex, and 3 for the Non Servo.

Fig. 4 is a graph showing the relationship of brake torque in inch-pounds to hydraulic line pressure in pounds per square inch for each of the three brake types under discussion. These curves were plotted from dynamometer tests made using brakes of the same size, the same lining material, wheel cylinders of the same diameter, and the same brake drum. This graph is shown solely to substantiate the theoretical effectiveness relationship of the three brake types, and is not to be interpreted as representing the ideal effectiveness of each. In combining variables to obtain the desired brake performance with each type, different brake lining material, for instance, is found to be desirable for one over another. Even though the linings are different, providing they are practical, the order of effectiveness is generally maintained.

In order to make brake recommendations for vehicles, it has been helpful to determine the relative effectiveness of one brake type with respect to another. For this purpose an empirical formula has been used. Brake effectiveness factor, K, represents the ratio of tangential force at the drum to the applying force on the brake shoes. Tangential force on the drum becomes brake torque divided by the drum radius, and force on the shoes becomes the product of hydraulic line pressure and wheel cylinder area or effectiveness factor K = (brake torque) ÷ (drum radius × hydraulic line pressure × wheel cylinder area). In the case of mechanically actuated brakes, force on the shoes is determined analytically or graphically considering camshaft torque and cam ratio.

Effectiveness factors are influenced by several variables, some of which are:

Anchor ratio (ratio of drum radius to the distance from the center of the brake to the anchor); brake lining coefficient of friction; brake shoe construction, i.e., rigid or flexible; surface temperature of the lining and drum; drum deflection; finish of the drum rubbing surface; and drum material.

Disk Brakes

By W. R. Rodger
Chrysler Corp.

IT IS generally recognized throughout the industry that current drum brake designs, particularly on the larger cars, are approaching or have reached their maximum capacity unless some radical changes are made to increase their ability to dissipate heat. Although the many improvements in brake design and materials have substantially increased the heat resistance of the drum type of brake, several other trends have offset to a great extent such brake im-

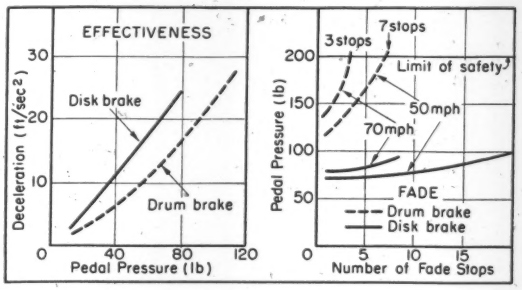


Fig. 7—Performance of experimental 11-inch disk brake compared with 2-cylinder drum brake of same diameter. Car weight was 3850 lb

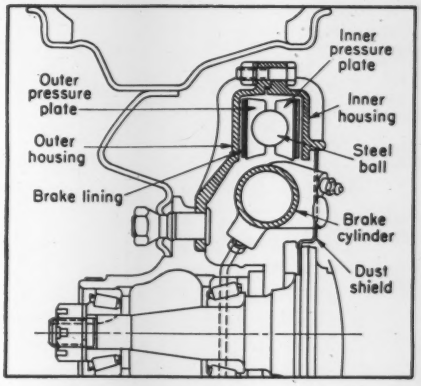


Fig. 8—Right — Cross section of Chrysler self-energizing disk brake

provements: average and maximum car speeds continue to increase, more engine power has placed an ever-increasing burden of heat rejection on the brake system, trend in car weights continues upward, etc. It is quite probable that improvements such as better brake linings and drums, forced air ventilation, and other design refinements will prolong the usefulness of the drum type brake, particularly on the smaller cars. However, the handwriting on the wall seems to indicate a trend toward disk brakes for passenger cars, a trail already blazed by fast aircraft and trains.

Study of the drum and disk type brakes will show that the latter provides three fundamental advantages over the drum brake which make it much more suitable for high heat absorption and dissipation:

1. Greatly reduced mechanical deflection by virtue of the structure of the disk and housing
2. Thermal expansion acts principally in a direction perpendicular to the plane of the applying forces, thereby minimizing the decreased performance caused by expansion in the drum brake
3. Large increase in braking swept-surface and in heat dissipating surface.

These three advantages result in improvements in brake operation which are fundamental; the brake is able to absorb and dissipate much greater energy without loss of braking efficiency. In addition, pedal effort may be reduced by higher self actuation, higher friction coefficient and higher mechanical advantage from pedal to wheels.

An early passenger car brake developed by the Goodyear Aircraft Corp. used the same principles that were successfully applied to its aircraft brake. In *Fig.* 5, a cross section of an experimental Goodyear design, it will be seen that a flat rotating disk is mounted on the wheel hub, and two round buttons of friction material are pushed against opposite sides of the disk through the action of a hydraulic piston and cylinder in a C-clamp construction. This brake is not self-energizing. The brake system makes use of high mechanical and hydraulic ratios and an automatic lining wear adjuster is provided.

This brake is unique because of the extremely low area of the friction material. In experimental models tested, the total lining area on a front brake was only 9.8 sq in., compared to 50.4 sq in. for a conventional 12-inch x 2-inch drum brake. A dense friction material with a low wear rate was used. By virtue of this low coverage of the disk by friction material, there is a large surface of the disk exposed for potential cooling. Thereby a low loss in brake output after repeated stopping is achieved. Two alternatives for mounting the brake may be chosen: allow the disk to float endwise on its mounting, with a fixed cylinder; or allow the cylinder to move endwise by means of a pantagraph support linkage, with a fixed disk.

The Budd Co. railroad brake is similar in principle to the Goodyear brake, but it is designed for even more rapid energy conversion. The rotating disk is hollow, with vanes to force air through the disk like a centrifugal blower, thus providing very effective forced-draft cooling. This brake also uses a resilient rubber backing for the linings to achieve more uniform normal force. As on the Goodyear aircraft brake, auxiliary power is used for brake application, inasmuch as this is a non-self-energizing design. This brake has been very successful in increasing railway braking performance and decreasing servicing.

Both the Milan-Kinmont and the American Chain and Cable disk brake designs have no self-energization and are therefore unsuitable for passenger cars.

The Auto Specialties Manufacturing Co. has been developing and building self-energizing Lambert disk brakes for tractors. This company has also developed several self-energizing disk brake designs for passenger cars. One of the earlier designs is shown in *Fig.* 6. The housing halves are made of aluminum, with gray iron friction liners cast into the aluminum. Two aluminum pressure plates, carrying the bonded friction material, are mounted on the axle and disposed between the inner friction surfaces of the housings. The plates are rotated relative to each other by two single-acting hydraulic cylinders, and are caused to separate and bear against the friction surfaces of the housings by the action of steel balls. which are forced to climb ramped pockets in the opposed plates. One pressure plate is fixed to the axle to transmit torque, and the rotation of the housing against the friction material on the non-anchored plate tends to further separate the plates, and thereby to increase the braking torque. To compensate for lining wear, this brake has a simple automatic adjuster, consisting of a washer bearing on a rod.

Self-energizing action of this brake provides ac-

ceptable light pedal effort for the heaviest cars, with good resistance to fade, loss of pedal, and erratic action under severe abuse. *Fig.* 7 shows the performance of an 11-inch experimental disk brake of this design, as compared to a 2-cylinder 11-inch drum brake. It will be noted that there is a 35 per cent reduction in pedal effort, together with a significant gain in fade resistance for the experimental brake. Compared to the more highly self-energized but less stable drum brakes, there is less of a gain in effectiveness; however, the disk brake pedal pressures are as light as the most effective drum brake of this type and up to 20 per cent lighter than others, depending upon whether the design emphasis was for maximum effectiveness or stability.

Chrysler Disk Brake Design

Fig. 8 shows a cross section of the Chrysler production disk brake. In this design the two-piece finned cast iron housing is mounted to the wheel hub. The pressure plates, which carry the bonded friction material, are mounted on a four-arm spider attached to the axle. Two single-acting cylinders are mounted on opposite sides of the inner pressure plate, and the piston push rods bear against lugs cast in the outer pressure plate, thus causing relative rotation of the plates when hydraulic pressure is applied. Six ⅞-inch diameter balls are located in ramped pockets generated in the pressure plates. As the hydraulic cylinders force relative rotation of the two pressure plates, the balls must climb the ramps and cause plate separation, which is further increased by the self-energizing action of the non-anchored pressure plate.

The inner pressure plates on the front brakes are anchored to the spiders for both forward and reverse rotation, the outer pressure plates being free to rotate. Therefore, the front brakes are self-energizing for forward motion only. On the rear brakes, the inner pressure plates anchor against the spider arms during forward braking to allow the outer pressure plates to be energized; in reverse, the outer pressure plates rotate around to stop against the spider arms and thus become the anchored plates to allow the inner plates to be energized. In reverse, then, only the rear brakes are self-energizing, but car speeds in reverse are so low that the somewhat higher pedal effort is of no consequence.

The wheel cylinder diameters are 1¼ inch for the front brakes, and 1 inch for the rear brakes, providing a brake distribution of 61 per cent front and 39 per cent rear. A 1-inch diameter master cylinder, with an 8.8:1 pedal ratio is used with this brake on a Chrysler Crown Imperial car. The performance of the production brake compared with the 12-inch drum brake is shown in *Fig.* 9.

The most significant single feature in this brake is its self-energization. This is accomplished very simply through the medium of the six steel balls, which are forced to climb the ramps in the pressure plate ball pockets as the plates are rotated relatively by the wheel cylinders. By changing the angle of ramp in the ball pockets, the self-energization can be varied, up or down, to accommodate any friction coefficient or service requirements. On the one hand, high self-energization is desirable in order to achieve low pedal effort; on the other hand, care must be taken to avoid too high a degree of self-energization in order to preclude over-sensitivity and self-locking.

The output vs. input relationships of the brake are of interest. Reference is made to the ball and ramp diagram and force polygon in *Fig.* 10. For a 35 degree ramp (experimental brake) and a friction coefficient of 0.35, the brake frictional force is twice cylinder force. The 32½-degree ramp in the production brake provides a brake frictional force which is 2.44 times cylinder force. In *Fig.* 10, P = wheel cylinder force, F = frictional force on one disk, N = normal force, R = resultant force, μ = coefficient of friction, ϕ = ramp angle, and F' = total frictional force on brake.

Forces are considered acting at the mean lining radius. From the frictional relationship,

$$F = \mu N, \text{ or } N = \frac{F}{\mu}$$

and from the force polygon in *Fig.* 10,

$$\tan \phi = \frac{P + F}{N}$$

$$N \tan \phi = P + F$$

$$\frac{F}{\mu} \tan \phi = P + F$$

$$F\left(\frac{\tan \phi}{\mu} - 1\right) = P$$

But since $F' = 2F$ for two disks

$$F' = \left(\frac{2}{\frac{\tan \phi}{\mu} - 1}\right) P$$

Brake designers use an *actuation constant* and an *actuation factor* to measure the degree of self-energizing in a brake. The actuation constant is a mathe-

(*Concluded on Page* 199)

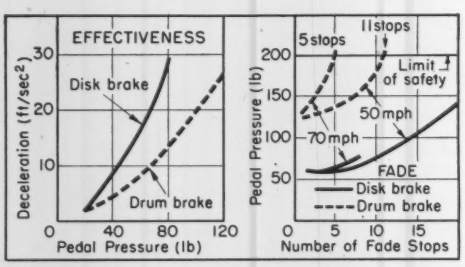
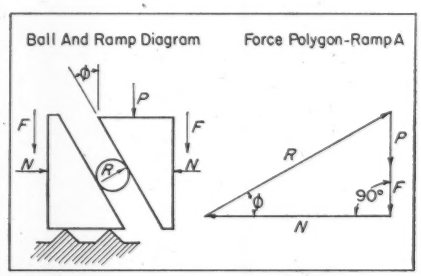


Fig. 9—Left — Performance comparison of 12-inch Chrysler disk brake with 12-inch, 2-cylinder drum brake. Car weight in these tests, 4300 lb

Fig. 10 — Right — Relation of forces in Chrysler disk brake



PRODUCTION AND DESIGN

Modern Practices in Manufacture

Charts Simplify Cost Calculations

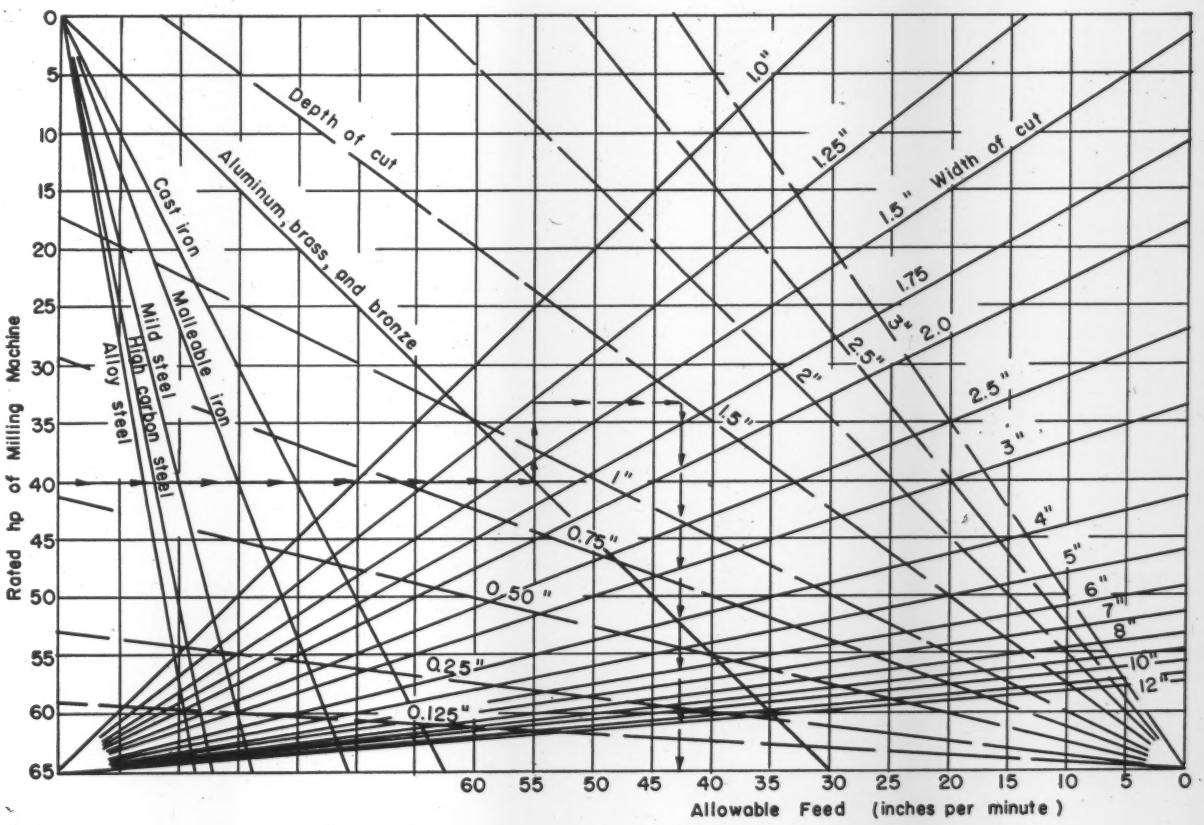
Part 4—Milling Feed and Time

By Tyler G. Hicks

Mechanical Engineer
Summit, N. Y.

CONCLUDING this series of charts covering major production operations, the two accompanying charts deal with milling machine feed and the subsequent milling time required for a specific operation.

MILLING MACHINE FEED: The knee and column



milling machines are in all probability the most common type in use today. Allowable cutter feed for these machines is a function of the horsepower available, metal cut, width of cut, and depth of cut. The chart on preceding page permits determination of feed in inches per minute.

Example: What is the allowable feed for a 40-horsepower milling machine when making a 1.250-inch wide by 1.500-inch deep cut in an aluminum member?

Enter the chart at 40 horsepower on the left and project horizontally to the proper material line, then vertically to the 1.250-inch width line and horizontally to the 1.500-inch depth line. At the bottom of the chart read the allowable feed as 42.6 inches per minute.

The construction of the chart on the preceding page is based on the following relationship:

$$F = \frac{P K}{w d}$$

where

F = Feed, inches per minute
P = Available machine power, horsepower
K = Metal milling constant
w = Width of cut, inches
d = Depth of cut, inches

The metal milling constants have been incorporated into the charts direct but can be found in various available handbooks.

MILLING MACHINE TIME: The time required to mill a given member is a function of the length of cut, the approach, overtravel, and feed. Allowable feed was found with the chart on the previous page. The chart shown on this page correlates feed with the remaining factors for typical milling problems.

Example: Determine the time required to mill a cast iron member when the length of cut is 8 inches, the approach to the piece 4 inches, the overtravel of the cutter ½-inch, and the allowable feed is 10 inches per minute.

Enter the chart at the bottom with a cut length of 8 inches and project vertically to the 4-inch approach line. From this point project across horizontally to the ½-inch overtravel line and then up vertically to the 10-inch feed line. At the right read 1.25 minutes milling time required.

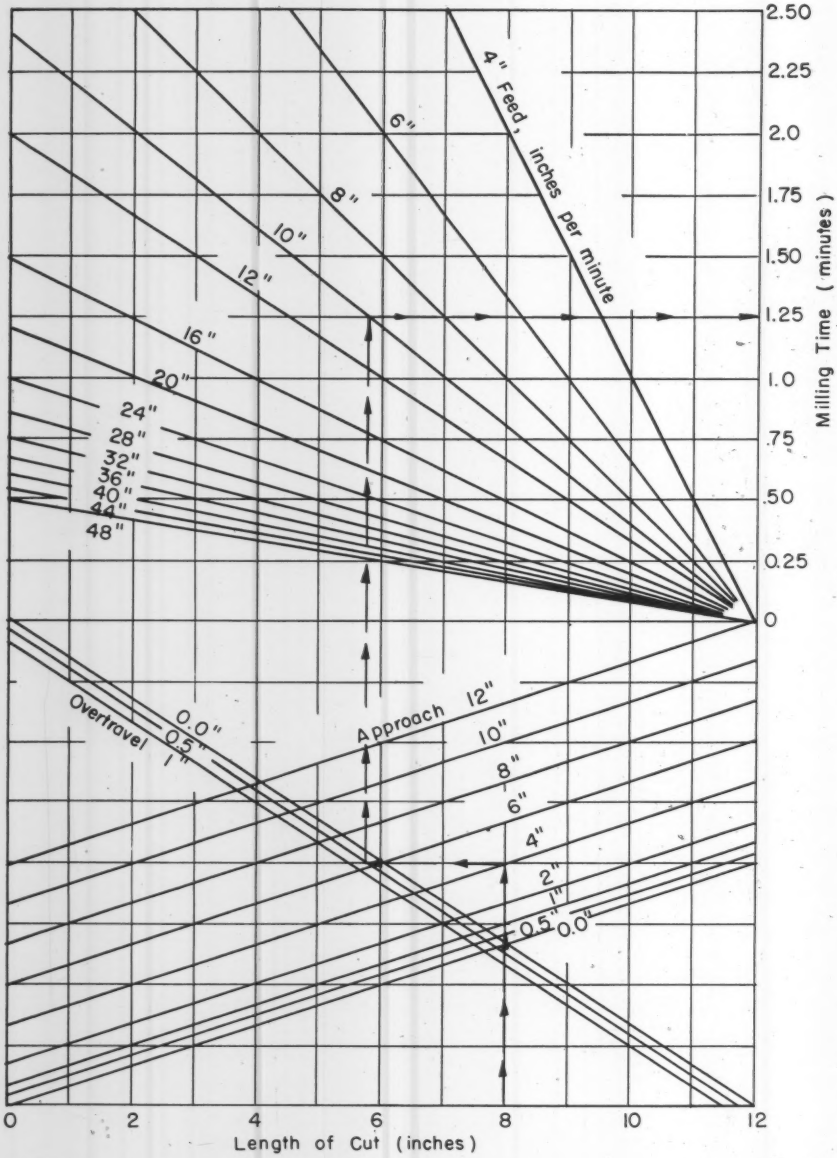
The chart is based on the relationship

$$M = \frac{(l + a + o)}{F}$$

where

M = Milling time, minutes
l = Length of cut, inches
a = Approach distance of the cutter, inches
o = Overtravel distance of the cutter, inches
F = Feed, inches per minute

CONCLUSIONS: Not only can charts such as those included in this series of six facilitate the determination of best procedures to employ for a given new part but can also be of considerable value in estimating work. Values obtained from the charts are sufficiently accurate to permit rough calculations on the machining cost of major parts prior to completion of final detailed drawings.



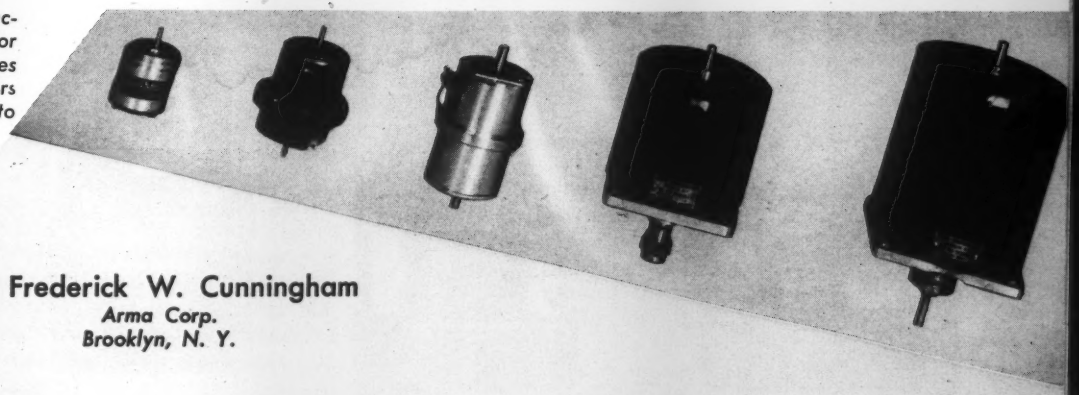


Fig. 17—Two-phase induction motors designed for use as servos. For voltages from 24 to 115, motors develop torques from 1 to 70 oz-in.

By Frederick W. Cunningham
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Brooklyn, N. Y.

Employing Computer Components in Machine Control

Part 3–Applications

CHARACTERISTICS of various electromechanical computing components have been discussed in the two previous articles of this series. This article, after a brief discussion of servo components, will consider some applications of computers for solving common problems.

Many operations of computation involve servomechanisms. As long as mere computation is the goal, the mechanisms can be very small and the motors can be controlled with one watt or so. When the final output of the computer is required to position something more than a dial, higher power servo motors and amplifiers are needed.

In servos having maximum power outputs up to fifteen watts, the two-phase induction motor with a cup rotor, a squirrel cage, or a solid rotor is satisfactory. Above this size the losses due to continuous operation at low speeds cause high temperature rises, and special means for cooling are required. Many motors have been equipped with separately driven cooling fans. Another approach to this problem is utilization of a wound-rotor induction motor, in which the stalled power is largely dissipated in external resistors. A further advantage of this construction is that the speed-torque curve can be varied within wide limits by use of a network in the rotor circuit. *Fig.* 17 shows some standard motors. Typical amplifiers for use with these motors are illustrated in *Fig.* 18.

When power is to be transmitted from one shaft to another which it does not intersect, the use of a pair of helical gears suggests itself. The determination of the helix angles of the gears is a little involved, and according to Buckingham, the result must be obtained by trial.

Equations for the helix angles are $\psi_1 + \psi_2 = \Sigma$ and

$$\frac{M}{\cos\psi_2} + \frac{1}{\cos\psi_1} = \frac{2CP_n}{N_1}$$

where ψ_1 and ψ_2 are the angles desired, Σ is the shaft angle, M is the gear ratio, C is the center distance, P_n is the normal pitch, and N_1 is the number of teeth on the driver. P_n, the normal pitch, is restricted to values for which cutters are available. By writing M as a fraction in its lowest terms, a/b, the number of teeth on the two gears are, respectively, an and bn and we have

$$an\cos\psi_1 + bn\cos\psi_2 = 2CP_n\,(\cos\psi_1\cos\psi_2)$$

where n is any integer greater than 0.

In *Fig.* 19 is shown an instrumental solution of

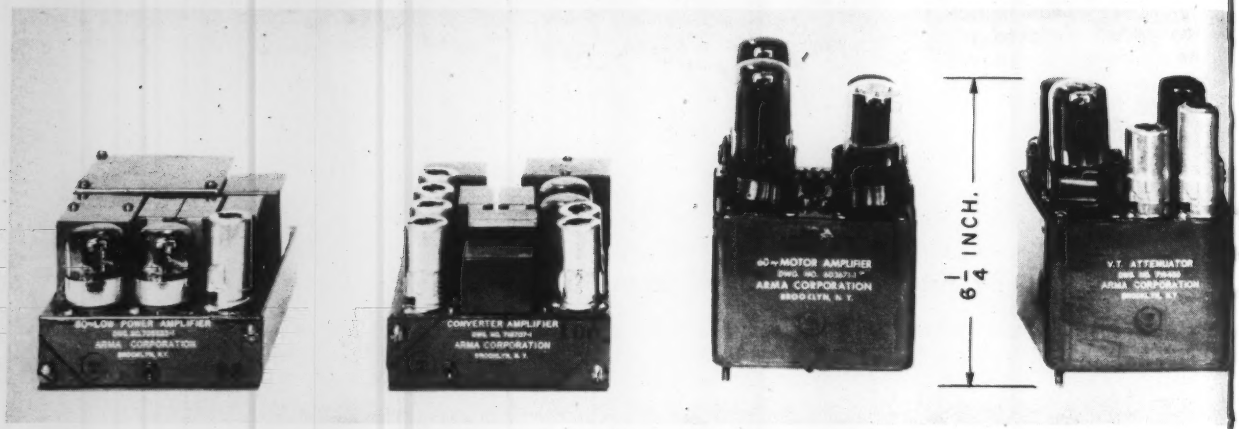


Fig. 18—Servo amplifiers designed for use with motors have power outputs up to 24 watts

the problem. Two resolvers are used for ψ_1 and two for ψ_2, each pair being geared together, and the two pairs being geared together through a differential. The third shaft of the differential is set at the shaft angle Σ.

The first resolver of each set is excited with voltages a and b, respectively. Since these are both integers, they may be provided by a tapped transformer. The sum of the outputs of these resolvers is fed to a tapped autotransformer. This is used because n is an integer and so must move in steps. The output of this transformer is then $an \cos \psi_1 + bn \cos \psi_2$. As $2P_n$ is determined by available tools and the tool to be selected is determined by the load to be carried, it also can best be obtained from a tapped transformer. This voltage is fed to an induction potentiometer to be multiplied by C. The voltage $2CP_n$ then excites the second ψ_1 resolver through a booster amplifier, and this resolver in turn excites the second ψ_2 resolver from its cosine output. The output of this resolver should be equal to $an \cos \psi_1 + bn \cos \psi_2$, so the difference between them is used to excite a servomechanism that positions the ψ_1 and ψ_2 resolvers.

The value of n can be selected at will from 1 up to a maximum possible value. In general, there are two sets of values of the helix angles which will yield a solution for a given value of n. If n is small it may be found that one of the helix angles will approach 90 degrees while the other will become negative. The preferred set of the two is that in which the helix angle of the driving gear is the greater. The best value of n is usually the largest that will give a solution in which both helix angles are positive.

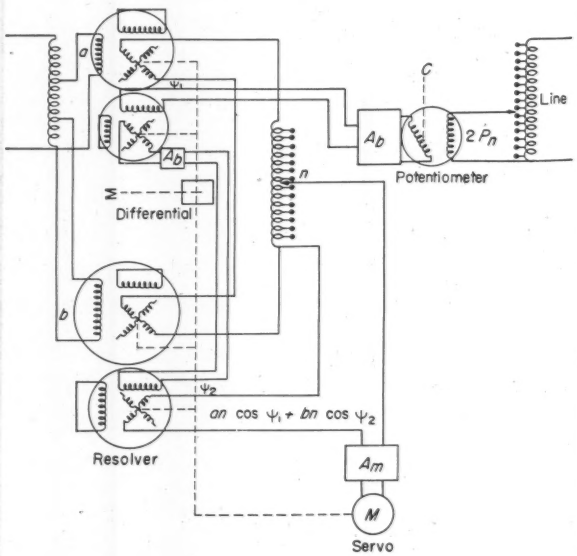
Computes and Controls Contour

The conventional method of cutting cams involves step by step computation of points followed by alternate movements in radius and rotation, with the continuous danger that the operator will make the adjustments in the wrong order. Then the bumpy result has to be smoothed by hand filing, with the further danger of taking off too much. A great deal of this can be avoided by automatic computation of a desired curve, with servo actuation of the cutting machine controls.

For example, it is often required that a harmonic rise be provided between two angles. A resolver can be driven by a servo so as to make a 180 degree rotation between the start and finish of the rise. Then its output can be used to control the position of the cutter. The amount of rise can be controlled simply by adjustment of the excitation of the resolver. *Fig.* 20 shows the essential components of such a system.

The rotary table of a vertical milling machine is geared to an induction potentiometer through a differential, of which the third member is used as a zero setter. The output of this potentiometer is fed through a second unit which is set to correspond to the angle through which the cam is to turn while it

Fig. 19—Circuit for determining the helix angles for a pair of helical gears



makes the rise. The output of this potentiometer is matched to that of a third unit by means of a servo. The excitation of this unit is such that it will travel from end to end while the cam is rotated through the desired angle.

Geared to this third potentiometer is a resolver that turns 180 degrees at the same time. This resolver is excited from a fourth induction potentiometer, which is set in accordance with the desired throw of the cam. The output of the resolver is matched to that of a fifth potentiometer which is geared to the longitudinal feed of the milling machine through a differential, the third member of which is provided with a clampable hand crank.

System Is Set to Zero Position

In operation, the cutter is first removed from the spindle and the rotary table turned to the angle at which the midpoint of the rise is to be. The zero setting crank is next adjusted and clamped so that the output of the first induction potentiometer is zero. This can be determined by observation of the servo which positions the resolver. The longitudinal or radial travel zero can now be set and clamped if desired, though this will probably be finally adjusted during the cutting operation. The rotary table is then turned to the angle at which the rise is to start or end and the cam angle potentiometer adjusted so the resolver will be just 90 degrees from its zero position. The resolver excitation is next adjusted so the resolver will be just 90 degrees from its zero position by means of the cam throw potentiometer so that the radial motion is correct in magnitude.

The cutter can now be replaced and the machine started. As the rotary table is driven, the radial feed will follow automatically. If desired, the radial zero setting can be backed off for roughing cuts.

Generates Involute Gear Teeth

Another computer that might be useful is one for generating involute gear teeth. The equation of an involute is $a = \sqrt{r^2 - 1} - \text{arc tan } \sqrt{r^2 - 1}$, where a is the angle measured from the start of the involute and r is the ratio of the radius to that of the base circle. In order that the electrical components of the computer be used over the maximum possible range it is advantageous to introduce the quantity d, which is $r-1$. Then $a = \sqrt{d(d+2)} - \text{arc tan } \sqrt{d(d+2)}$. If a machine is to be controlled to cut an involute, it is convenient to determine d in terms of a, rather than a in terms of d. An instrumentation for this is shown in *Fig.* 21.

If potentiometers A and B are rotated through an angle d, the output of A will be d, assuming proper scaling. To d is added 2, and this is used to excite potentiometer B. Then the output of this potentiometer is $d(d+2)$. Two other potentiometers C and D are geared together, with D excited from C and C excited from the line. If the output of D is the same as that of B the potentiometers have been rotated through an angle $\sqrt{d(d+2)}$.

A resolver F has one primary excited by a unit voltage and the other by the output of potentiometer C. A servo drives this resolver to a null output of one coil. Then since the output $\sqrt{d(d+2)} \cos H - \sin H = 0$, $\tan H = \sqrt{d(d+2)}$ and $H = \text{arc tan } \sqrt{d(d+2)}$. The rotation of the resolver is subtracted from that of the potentiometers C and D by means of a differential to

Fig. 20—Machine control for generating a harmonic rise for a cam

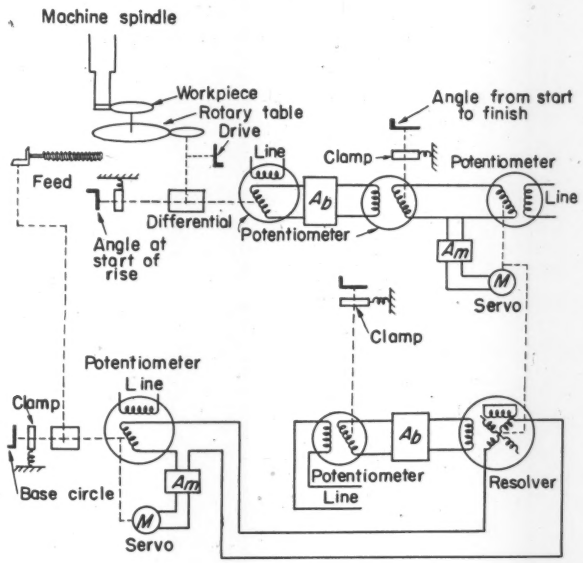
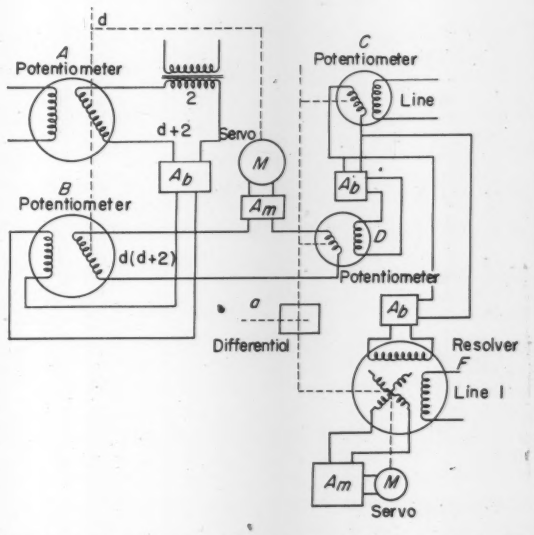


Fig. 21—Control schematic for generating involute gear teeth



give angle a. Since it is desired to find d, the servo which matches the output of potentiometer B to that of D is used to drive A and B. Then as input a is changed, d will follow. If it is desired to find a in terms of d, the servo will be used to drive C and D. In the former case, the output of potentiometer A can be transmitted to a distant point and fed through a tapped transformer to control a servo on a cutter to form an involute on any desired base circle.

It was mentioned that it is advantageous to introduce the quantity $d = r - 1$ in order to use the potentiometers over the greatest possible range. Ordinarily d will not be over 0.25 and it becomes possible to use the full range of the d potentiometers to represent this value. If they are used for r, since this varies from 1 to 1.25, only 1/5 the range of the potentiometers will be covered, and 5 times as much error from this source can be expected. This is a point of great importance in analogue computers.

A more generally useful computer is one that will generate a curve of any form desired. This can be done by a process of continuous interpolation between given points. This interpolation may be of any order, but the third order will ordinarily be adequate. Such a computer has the advantage that its accuracy is not limited by the components, as they are used only to represent successive differences between the given points under consideration.

Simplifying Offset Crank Analysis

By George J. Frey
Ohio Mechanics Institute
Cincinnati, O.

MECHANISM problems occasionally involve linkages in which the center of rotation of a crank is offset from the line of action of the crosshead. The simple mathematical solution shown here is not subject to the limitations of graphical, trial-and-error solutions.

Knowing the length of crosshead stroke, a, length of connecting rod, l, and the offset of the crank center, h, the problem is to solve for the radius of the crank, r, and to locate the crank center. From the sketch, $c = l - r$, and $b = l + r$. The area of the triangle, abc, is

$$\text{Area} = \frac{1}{2}\,ah = \sqrt{s\,(s-a)\,(s-b)\,(s-c)} \quad \ldots\ldots (1)$$

where $2s = a + b + c = a + 2l$. Squaring both sides of Equation 1 and substituting $s = a/2 + l$, $b = l + r$ and $c = l - r$ gives

$$r^2 = \frac{a^2}{4} = \frac{a^2h^2}{4l^2 - a^2}$$

$$= \frac{a^2}{4}\left(1 - \frac{h^2}{l^2 - \frac{a^2}{4}}\right)$$

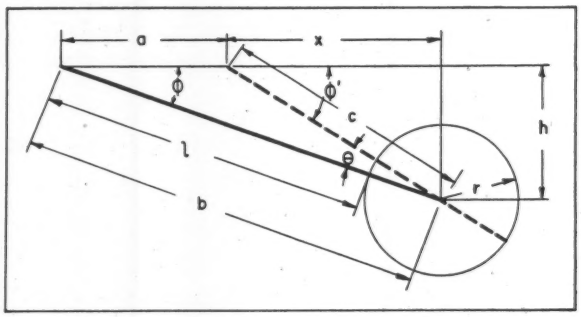
Then, simplifying by taking the square root gives

$$r = \frac{a}{2}\sqrt{1 - \frac{h^2}{l^2 - \left(\frac{a}{2}\right)^2}} \quad \ldots\ldots (2)$$

Also, from the sketch, simple trigonometry gives

$$\phi' = \sin^{-1}\frac{h}{l - r}$$

$$x = (l - r)\cos\phi' \quad \ldots\ldots (3)$$



which locates the crank center. Also

$$\phi = \sin^{-1}\frac{h}{l + r}$$

$$\theta = \phi' - \phi$$

One stroke occurs during $180 + \theta$ degrees and the other during $180 - \theta$ degrees.

If crank radius, r, is known, Equation 2 can be rearranged as

$$h = \sqrt{\left(\frac{a^2}{4} - r^2\right)\left(\frac{4l^2}{a^2} - 1\right)} \quad \ldots\ldots (4)$$

or if connecting rod length, l, is the unknown, then

$$l = \frac{a}{2}\sqrt{\frac{4h^2}{a^2 - 4r^2} + 1} \quad \ldots\ldots (5)$$

They Say...

"Mechanical and technical advancements in the textile industry have introduced complications with which the average machine operator cannot cope. Engineers are needed in the textile industry now more than ever before because mechanical complications have eliminated the individual who can qualify as both operator and maintenance man" — JAMES R. LONGSTREET, *The Warner & Swasey Co.*

Gear Design for Strength and Wear

Gear Beam Strength

By F. L. Lindemuth
Chief Engineer
William B. Pollock Co.
Youngstown, Ohio

SATISFYING the requirements for beam strength of gear teeth is often a trial-and-error process during the calculation of a gear train. The chart on the following page has been constructed as a means of reducing the tedious detail of the usual computational procedure. It provides a rapid method of applying the Lewis equation in determining satisfactory relationships of such factors as number of teeth, pitch, power and speed.

Power transmitted by a gear is simply the product of the load on a tooth and the pitch-line speed. The Lewis equation treats a gear tooth as a cantilever beam and expresses the safe load as

$$W = spFy \quad \ldots\ldots (1)$$

where W = safe static beam load on teeth, lb; s = safe static bending stress for material, psi; p = circular pitch, inches; F = effective face width, inches; and y = Lewis tooth form factor.

For practical applications,

$$W = K_1 W_k \quad \ldots\ldots (2)$$

where K_1 = combined dynamic load and service factor and W_k = working load, lb.

Chart method of solution is based upon a face width four times the circular pitch:

$$F = 4p = \frac{4\pi}{P_d} \quad \ldots\ldots (3)$$

where P_d = diametral pitch. Thus, for other proportions a face adjustment factor is required:

$$K_2 = \frac{F/p}{4} = \frac{F P_d}{4\pi} \quad \ldots\ldots (4)$$

Combining Equations 1 through 4 and solving for working load,

$$W_k = \frac{4K_2 sy\pi^2}{K_1 P_d{}^2} \quad \ldots\ldots (5)$$

Two well known equations introduce other required properties:

$$W_k = \frac{33000\,P}{v} \quad \ldots\ldots (6)$$

$$v = \frac{\pi n N_P}{12P_d} \quad \ldots\ldots (7)$$

where P = horsepower; v = pitch line velocity, fpm; n = pinion speed, rpm; and N_P = number of pinion teeth.

Combining Equations 5, 6 and 7

$$\frac{4K_2\,s\,y\,\pi^2}{K_1\,P_d{}^2} = \frac{33000\,P}{v} = \frac{396000P\,P_d}{\pi n N_P} \quad \ldots\ldots (8)$$

Fig. 1 is a composite graph of these relationships.

Three involute forms are built into the chart: 14½-degree full, 20-degree full, and 20-degree stub. To simplify dealing with the Lewis form factor, an averaging method has been employed. According to the chart, for example, the tooth-strength ratio of a 50-tooth gear to a 15-tooth gear is the same in the 14½-degree full and the 20-degree stub systems. Theoretically this is not true, but the difference is negligible in practical design.

Use of the chart is easily explained by an example. Given: horsepower P = 100; pinion speed n = 500 rpm; pinion teeth N_P = 30; tooth form = 20-degree full; face width $F = 3p$; stress s = 80,000 psi; estimated dynamic load and service factor K_1 = 2. Find: diametral pitch P_d, pitch line speed v, and effective face width F.

The procedure is as follows:

1. Obtain factor $K_2 s/K_1$. By Equation (4), K_2 = ¾. $K_2 s/K_1$ = (3)(80,000)/(4)(2) = 30,000
2. Obtain adjustment length a from small graph at left of chart. It is the distance along the 20-degree-full vertical from the 10,000 horizontal to the 30,000 slope
3. Along the horsepower scale, mark off a upward from the 100-hp point. From the upper point, proceed horizontally to the 30-tooth vertical and

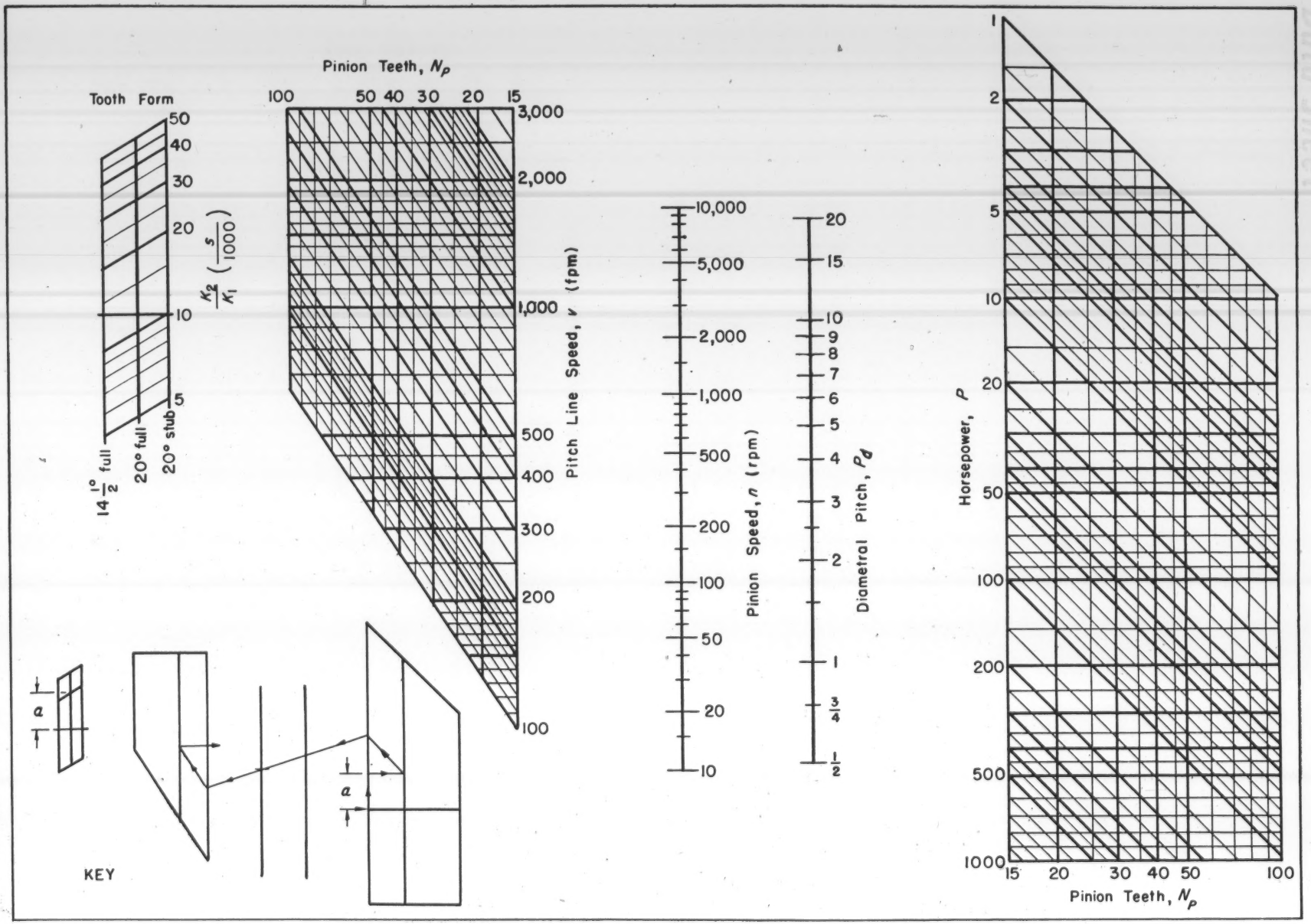

Tooth Form
14 1/2° full
20° full
20° stub
50
40
30
20
10
5
K2/K1 (s/1000)
Pinion Teeth, Np
100
50 40 30 20 15
3,000
2,000
1,000
500
400
300
200
100
Pitch Line Speed, v (fpm)
10,000
5,000
2,000
1,000
500
200
100
50
20
10
Pinion Speed, n (rpm)
20
15
10
9
8
7
6
5
4
3
2
1
3/4
1/2
Diametral Pitch, Pd
1
2
5
10
20
50
100
200
500
1000
Horsepower, P
15 20 30 40 50 100
Pinion Teeth, Np
a
a
KEY

Fig. 1—Left—By means of this chart, gear properties satisfying beam strength requirements may be determined quickly. Lewis factors are built into the grids for tooth form and horsepower versus pinion teeth

return to the left limit, paralleling the sloped lines

4. By a straight line join final point determined on horsepower axis in Step 3 and 500 point on rpm scale. Diametral pitch $P_d = 5.5$
5. Continue the same line to the 15-tooth vertical of the pitch-line speed grid. From point of intersection, parallel the sloped lines to the 30-tooth vertical and return horizontally to the pitch-line speed scale. Pitch line speed $v = 700$ fpm
6. Rearranging Equation 4, solve for effective face width: $F = (4\pi K_s)/P_d = (3\pi)/5.5 = 1.714$ inches.

Now known, the pitch-line speed can be checked against the expected gear quality and the dynamic load can be definitely ascertained. If requirements are not mutually satisfied by first trial, successive calculations may be made quickly with the chart.

Gear Wear Load

By W. R. Leopold
Special Products Div.
M. W. Kellogg Co.
Jersey City, N. J.

PREVENTING excessive wear of gear teeth requires selection of gear properties that provide safe contact stress. The accompanying charts offer a convenient method of satisfying this important design criterion for spur pinions meshing with external or internal gears.

Calculations are based upon the Hertz formula for maximum contact stress between two elastic cylinders:

$$s^2 = 0.35\, w_n \frac{\frac{1}{\rho_P} + \frac{1}{\rho_G}}{\frac{1}{E_P} + \frac{1}{E_G}} \quad \ldots\ldots (1)$$

where s = maximum contact stress, psi; w_n = normal load per unit length of contact, lb per in., ρ_P and ρ_G = radii of contacting contours, inches; and E_P and E_G = moduli of elasticity of contacting members, psi.

From the geometry of the involute and with contact stress taken at the pitch circles, the Hertz formula can be developed into the following special case for a spur pinion and an external gear:

$$s^2 = \frac{0.35 w_t}{\sin 2\phi} \left(\frac{\frac{1}{D_P} + \frac{1}{D_G}}{\frac{1}{E_P} + \frac{1}{E_G}} \right) \quad \ldots\ldots (2)$$

where w_t = tangential load per unit width of face, lb per in.; D_P = pitch diameter of spur pinion, inches; D_G = pitch diameter of external gear, inches; and ϕ = pressure angle, degrees.

For the graphical method presented here E_P and E_G are assumed to be equal. Hence, substituting E for E_P and E_G and rearranging,

$$w_t = \frac{s^2 \sin 2\phi}{0.70\, E} \left(\frac{D_P D_G}{D_P + D_G} \right) \quad \ldots\ldots (3)$$

For an internal gear and spur pinion, the following similar form can be derived:

$$w_t = \frac{s^2 \sin 2\phi}{0.70\, E} \left(\frac{D_P D_G}{D_G - D_P} \right) \quad \ldots\ldots (4)$$

Effective face width is introduced by the relationship:

$$F = \frac{W_t}{w_t} \quad \ldots\ldots (5)$$

where F = effective face width, inches, and W_t = total tangential load, lb.

In this graphical method, F is the variable to be calculated from W_t and w_t. *Figs.* 2 and 3 provide rapid means for determining the latter two factors. To find W_t the procedure is:

1. Calculate the tangential load acting at the pitch circles of the pair in question. Number of pinions sharing the total transmitted load, power, and dynamic loading should be taken into account
2. For the number of pinions sharing the load, and number of teeth on a pinion, obtain factor K_P* from *Fig.* 2
3. Multiply the tangential load from Step 1 by K_P to obtain the total design load W_t

To find w_t the procedure is:

4. Joining the pitch diameters D_P and D_G for the two gears by a straight line in *Fig.* 3, obtain K_w

Factor K_w is equal to w_t for the following conditions: $E = 30{,}000{,}000$ psi, $\phi = 20$ degrees, and $s = 100{,}000$ psi. Generally, no adjustment will be required for E if steel gears and pinions are used, but compensations must be made if different pressure angles and contact stress values are used.

5. Obtain pressure-angle factor K_ϕ from the conversion scale in *Fig.* 3
6. Obtain contact-stress factor K_s from the conversion scale in *Fig.* 3
7. Calculate unit design load by the equation

$$w_t = K_w\, K_\phi\, K_s \quad \ldots\ldots (6)$$

To find F, the face width,

8. Divide the total design load W_t (Step 3) by the unit design load w_t (Step 7), as indicated in Equation 5.

EXAMPLE: Find the safe effective face width for the following external-gear conditions: Number of pinions = 4, diametral pitch = 10, pitch diameter $D_P = 6.5$ inches, pitch diameter $D_G = 8.5$ inches, pressure angle $\phi = 14\frac{1}{2}$ degrees, limiting contact stress $s = 135{,}000$ psi, and tangential pitch line load = 1750 lb (7000 lb total for four pinions).

From *Fig.* 2, load compensation factor K_P is 0.74

*Based on data supplied by Wright Aeronautical Corp.

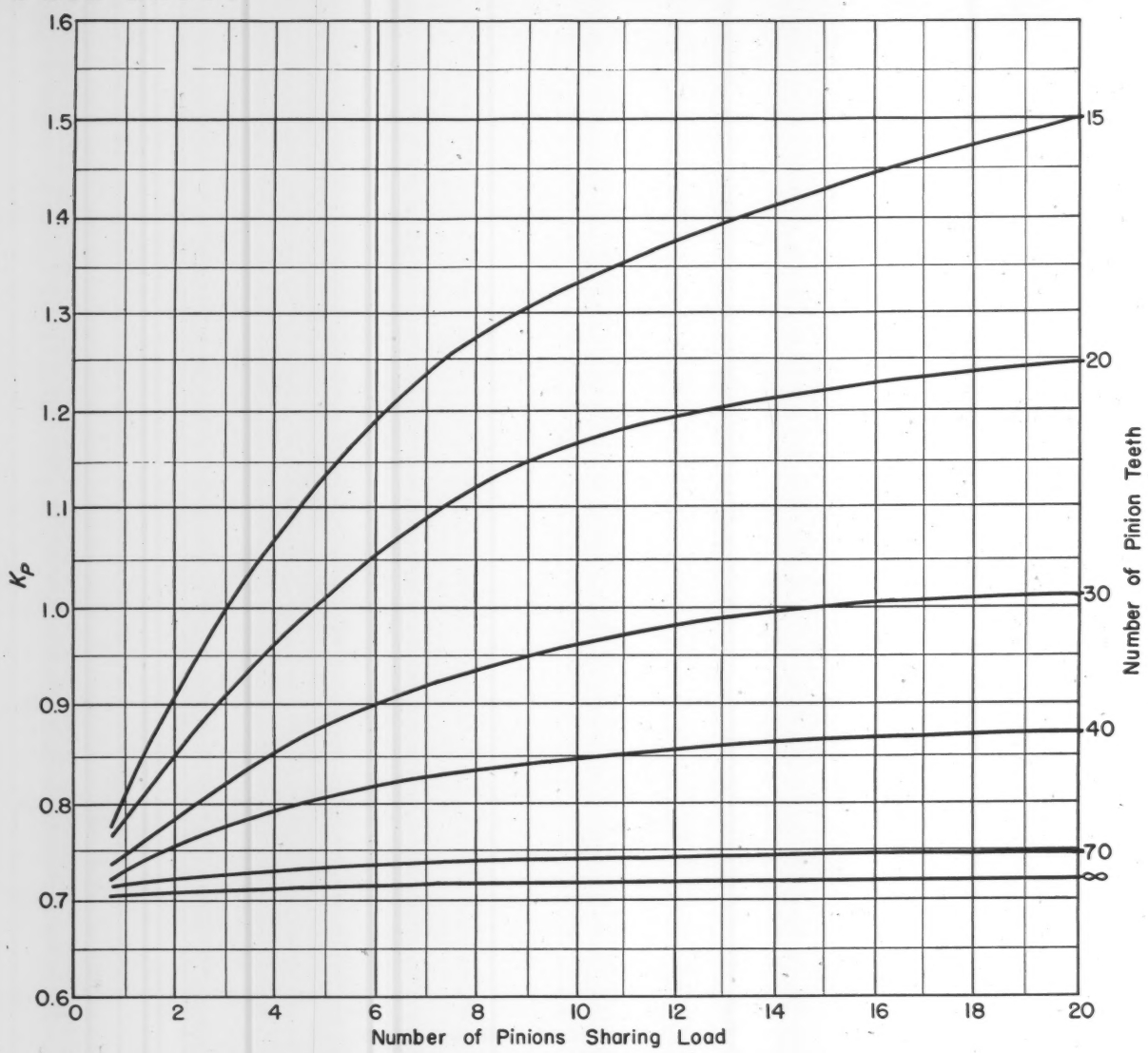


Fig. 2—Total limiting load for wear is influenced by number of pinions sharing the transmitted load and number of teeth on a pinion . Multiplying the load per pinion calculated from the expected service conditions by the factor plotted in this graph gives the total design load

for four 65-tooth pinions. Hence the total design load is $W_t = (0.74)\ (1750) = 1295$ lb.

From *Fig.* 3, $K_w = 1125$, $K_\phi = 0.75$ and $K_s = 1.82$. Therefore the unit design load is $w_t = (1125)\ (0.75)\ (1.82) = 1535$ lb.

Effective face width is the total load divided by the load per inch (Equation 5): $F = 1295/1535 = 0.844$ inch.

Scales for K_ϕ and K_s represent the relationships:

$$K_\phi = \frac{\sin 2\phi}{\sin 40^\circ}$$

$$K_s = \left(\frac{s}{100{,}000}\right)^2$$

The alignment chart for K_w represents the following relationship which is derived from Equation 3:

$$C\frac{1}{K_w} = \frac{1}{D_G} + \frac{1}{D_P}$$

where $C = (s^2 \sin 2\phi)/(0.70\ E)$ and $K_w = w_t$ only for arbitrarily assigned values of E, ϕ and s. The layout of the chart is quite simple. Equal linear scales are used for the vertical D_G axes and the horizontal D_P axis. The axis of K_w is inclined at 45 degrees in the quadrant bounded by D_G for external gears and D_P. Values of K_w are laid off according to the equation:

$$d = \sqrt{2}\frac{K_w}{C}$$

where d is the distance from the apparent origin along the axis of K_w and the scale of K_w is the same as of D_G and D_P.

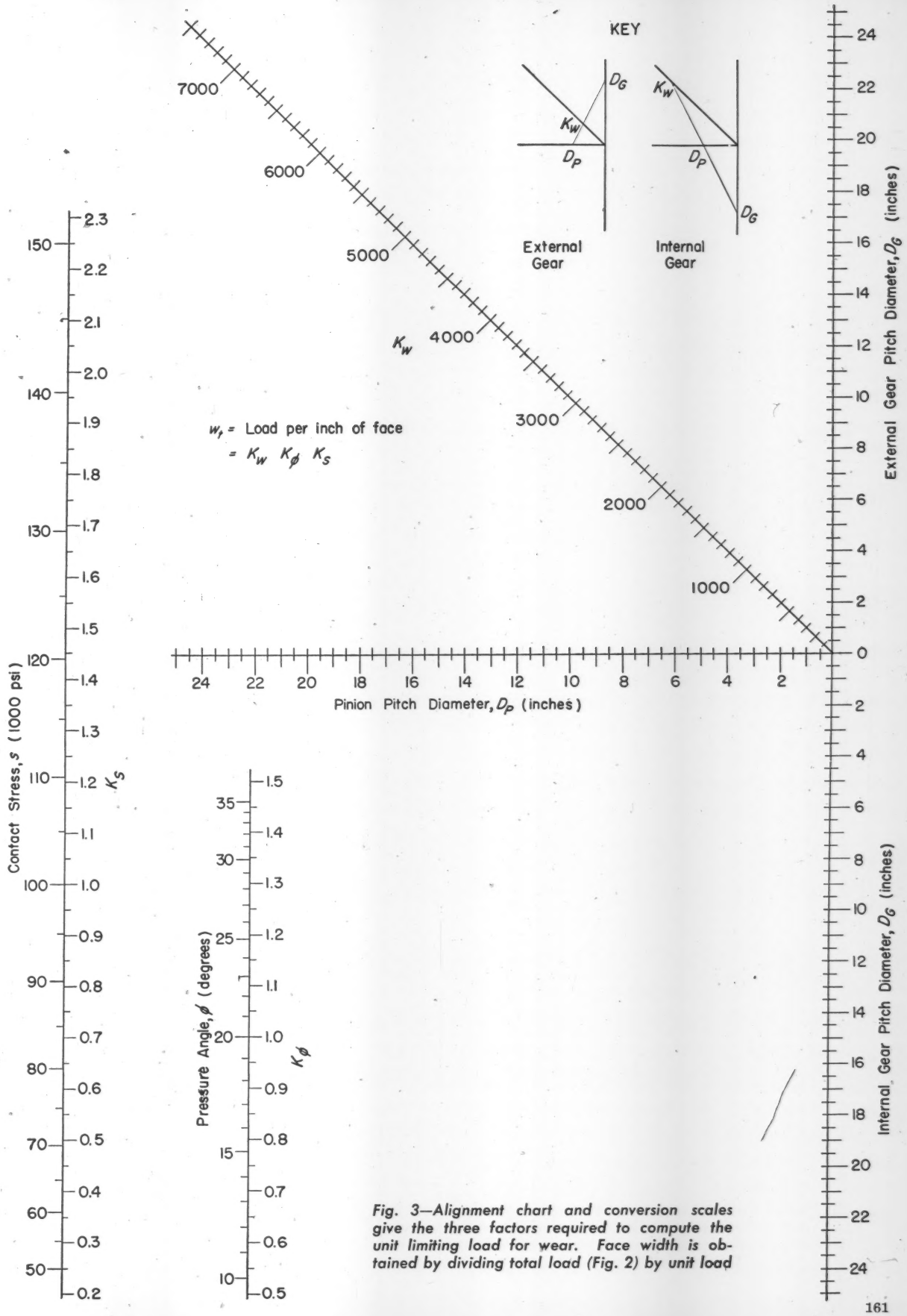


Fig. 3—Alignment chart and conversion scales give the three factors required to compute the unit limiting load for wear. Face width is obtained by dividing total load (Fig. 2) by unit load

MEN OF MACHINES

Southwest Research Institute has announced the appointment of **C. Desmond Pengelley** as chairman of engineering mechanics for the Institute. Formerly associated with the Curtiss-Wright Corp. as head of the flutter section dealing with flutter vibration and aeroelasticity, he was also engaged with aerodynamics, static testing, stress analysis and landing gear layout. Mr. Pengelley is credited with successfully designing and testing a two-dimensional parameter, typical of actual supersonic flying machines.

♦

Carl E. Schmitz, vice president and director of engineering of Crane Packing Co., was elected vice-president-at-large of the American Society of Lubrication Engineers at the annual convention. His other activities include chairmanship of a subcommittee on engine design for the SAE and membership on the conference committee of the National Conference of Industrial Hydraulics. He has earned national recognition as an authority on design and application of mechanical seals and their uses in connection with lubricating problems.

♦

Appointment of **Frank W. Curtis** as chief engineer of the entire Machine Tool Division which includes milling machines, oscillating grinders and the Van Norman-Fitchburg line of production grinding machines has been announced by the Van Norman Co. Mr. Curtis was president of ASTE in 1941-42. For the past six years, he has been consulting specialist on high-frequency and dielectric heating and prior to that had been associated with the Van Norman Co.

♦

A. J. Maslin has been appointed assistant to the engineering manager Transformer Division of the Westinghouse Electric Corp., at Sharon, Pa. In his new assignment, Mr. Maslin will continue to head the general transformer development section and in addition will be responsible for insulation development and co-ordination of many other engineering projects at the plant. In 1947, he was appointed advisory engineer for the Transformer Division, working on special theoretical projects such as analysis of abnormal voltages appearing across preventive autotransformers during under load tap changing and development of apparatus and circuits for use with mercury-arc rectifiers. Mr. Maslin has written many papers and articles for professional publications and technical magazines and has been granted nine patents mainly on transformer developments.

C. Desmond Pengelley

Carl E. Schmitz

Frank W. Curtis

Thomas C. Searle was appointed chief of the drafting and design branch of the Electro Mechanical Laboratory of the Air Force Cambridge Research Laboratories. A graduate of the University of Tennessee with a B. S. degree, he attended Harvard Graduate School of Engineering and received his Masters' Degree in mechanical engineering. During the war, Mr. Searle served as a member of the U. S. Navy where he was an ordnance specialist and participated in the developing and installation of synthetic devices for the training of antiaircraft gunners.

Thomas C. Searle

♦

Appointment of **William A. Mussen** as supervisor of the electronics laboratory at Southwest Research Institute has been announced. He has been responsible for various phases of radio-proximity fuze development and instrumentation in the U. S. Naval Ordnance Laboratory. Mr. Mussen is also possessor of a Navy Bureau of Ordnance Development Award and also has had experience in exploration geophysics and railroad electrification.

♦

William W. McDowell was appointed to head all engineering and research of International Business Machines Corp., with headquarters in New York.

♦

Worthington Pump and Machinery Corp., announces that **Mr. Charles A. Butcher** has joined the company as assistant to the president. Mr. Butcher formerly was general manager of Crocker-Wheeler Electric Mfg. Co. and vice president of both the Joshua Hendy Corp. and Elliott Co. He is a member of the advisory committee of motor and generator section of NEMA. Mr. Butcher also is a Fellow of AIEE and an Associate of AISE.

♦

Farrel-Birmingham Co. Inc. announces the appointment of **Carl F. Schnuck** as director of engineering and **Warren C. Whittum** as chief engineer.

♦

Automatic Electric Co. announces that **Kenneth W. Graybill,** who has been chief engineer for the past twelve years, has been elected vice president. As vice president and chief engineer, Mr. Graybill will continue to give executive direction to the company's research, development and patent affairs, and to have staff supervision over manufacturing operations. In his work as chief engineer, he has become known as an authority on automatic switching techniques and also for his work in the broader fields of communication and electrical control. In December, 1948, he received the Talbot G. Martin Award of the Strowger Automatic Club for his outstanding contributions to the art and science of automatic telephony.

♦

Frank Gimblette, manager of the General Electric Company's Taunton, Mass., plant since 1947, has been appointed to the Plastics Division's manufacturing headquarters staff. Succeeding Mr. Gimblette will be **Thomas H. Way.**

♦

Arthur P. Shanklin, vice president of Carrier Corp., was elected president of the Air Conditioning and Refrigerating Machinery Association at the annual meeting.

♦

Election of **Cecil H. Gay,** works manager of the Barberton, Ohio plant of Babcock & Wilcox Co., as vice president has been announced.

♦

Robert B. Moore has been appointed manager of the mining division of General Electric's industrial engineering divisions. He will succeed **R. S. Sage** who plans to retire.

♦

George H. Freers has been promoted from the position of chief engineer to vice president in charge of engineering of Marmon-Herrington Co. Inc., Indianapolis.

♦

David L. Williams, who has been associated with the Bausch & Lomb Optical Co. for 13 years, has joined the Westinghouse Lamp Division, Bloomfield, N. J. Mr. Williams will specialize in reflection systems such as those used in automotive and locomotive headlights, and lighting for motion picture projectors. Some of this activity will be an expansion of work he had started at Bausch & Lomb.

♦

General Electric Co. announces the appointment of **Clarence H. Linder** as manager of engineering and acting manager of manufacturing of the company's Affiliated Manufacturing Companies' Department. Mr. Linder is a graduate of the University of Texas.

NEWS
OF MANUFACTURERS

The Babcock & Wilcox Tube Co., Beaver Falls, Pa., announce plans for entering limited fabrication of seamless and welded tubing in addition to its primary work of manufacturing such tubing. The types of fabrication to be handled under the new policy will include bending; end forming such as swaging, upsetting, flaring and expanding; flash-butt-welding for safe-ending and making long length tubing; and the production of tubing with various cross section shapes and other specialty tube work.

♦

Consolidated Engineering Corp., Pasadena, Calif., plans the construction of an up-to-date scientific instrument plant. Designed to provide maximum convenience and flexibility for the development and manufacture of precision instruments such as recording oscillographs and analytical mass spectrometers, the 66,000-sq ft, half-million dollar structure will house the administrative, engineering, research, and production departments of the company.

♦

A new company, the **Paramount Compressor Corp.,** has been organized in Chicago, Ill., to take over the manufacture of a newly improved all-purpose air compressor formerly made by the **Compressor Corp. of America,** St. Joseph, Mich. The new firm's plant and office are located at 512 North Sangamon St., Chicago 22, Ill.

♦

Because of the conflicting tradenames of other manufacturers, **The Engineering Mfg. Co.,** Sheboygan, Wis., manufacturer of drafting and navigating equipment, has changed its tradename to "Mayline". Products of the company were formerly known under the tradenames "Seco" and "Esco". The trademark of the man at the drafting board has been retained.

♦

The Watertown Mfg. Co., Watertown, Conn., has acquired the physical assets of the **Moulded Metals Co. Inc.,** also of Watertown. Existing facilities of the powdered metallurgy division have been augmented to handle exceptionally large work in gears, cams, pole pieces and other applications.

♦

A. O. Smith Corp., Milwaukee, Wis., is expanding its electric motor manufacturing facilities in the east. The expansion is to be made at Tipp City, O., where A. O. Smith recently bought **Whirl-A-Way Motors Inc.** The Tipp City operation is now the eastern motor division of Smith. A new 60,000-sq ft plant is to be built this summer with production expected to be underway in September. The line of fractional-horsepower motors manufactured at Tipp City will be expanded to include polyphase motors as well as the present line of single phase units.

♦

Hapman Conveyors Inc., Detroit, Mich., and **The C. H. Dutton Co.,** Kalamazoo, Mich., have merged. The new company will be known as the **Hapman-Dutton Co.** and will consist of two divisions, the Hapman Conveyors Division and the Dutton Boiler Division, with principal offices located at Detroit and Kalamazoo respectively.

♦

The DeVilbiss Co., Toledo, O., manufacturer of air compressor outfits, spray painting equipment and atomizers, is in the process of establishing a modern manufacturing plant in Somerset, Pa. Closer cooperation with suppliers and more rapid and economical distribution to customers are objectives of the expansion.

♦

The **Dryomatic Corp. of America** has been reorganized as a Virginia corporation. Plant and main offices have been moved from Baltimore, Md., to Alexandria, Va. The reorganized firm will be known as the **Dryomatic Corp.** and will continue the manufacture of dehumidifying units on an expanded production schedule.

♦

Electric Steel Foundry, 2141 NW 25th Ave., Portland, Ore., has added two furnaces, Moore "PT" and "ST" types, to its melting equipment. This represents the initial step in a plant-wide efficiency program designed to increase the production capacity of ESCO alloy steels.

♦

Gar Wood Industries Inc., Wayne, Mich., manufacturer of large truck equipment, has purchased the **National Truck Equipment Co.,** Waukesha, Wis. This property will be owned and operated by a newly formed and wholly owned subsidiary of Gar Wood to be known as the **National Lift Co.**

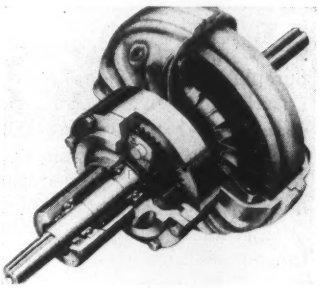
♦

Fairbanks-Morse & Co., Chicago, Ill., and **The Canadian Fairbanks-Morse Co. Ltd.,** Montreal, Canada, have entered the Canadian railroad diesel locomotive field through the purchase of substantial stock in the **Canadian Locomotive Co.,** Kingston, Ontario. The opposed-piston type diesel designed by Fairbanks-Morse at Beloit, Wis., will be the locomotive power source.

NEW PARTS AND MATERIALS

For additional information on these new developments see Page 179

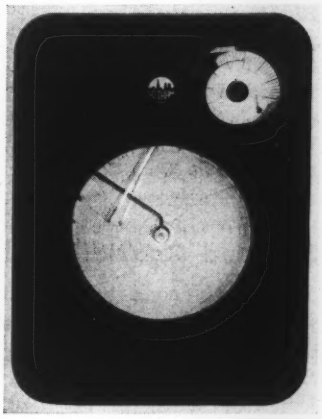
Hydraulic Coupler Drive



Predetermined lineal speed and constant tension are automatically maintained by this Hydro-Wynd unit. The new drive combines a hydraulic coupling with a planetary gear set, whose ring gear is fixed to the impeller or pump of the hydraulic coupling. The planet gear carrier is fixed to the runner of the coupling; the sun gear is fixed to the output shaft. In operation, automatic slip of the coupling reacts with the gear set against varying torque and speed requirements of the winding spool and changes output torque in direct proportion to load imposed. A variable speed drive can be used with this unit to control tension where frequent changes in load and speed are necessary. Manufacturer: Twin Disc Clutch Co., Racine, Wis.

For additional information circle MD 1 on Page 179

Program Recorder-Controller

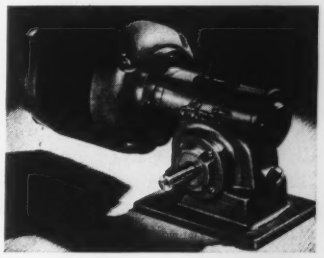


Standard cams used in this program recorder-controller provide a means for changing the rate rise, length of hold and holding temperature or pressure without cutting a cam for each cycle or inserting different cams. The holding temperature is changed by turning the control temperature knob.

Instrument has a 9-in. chart and is housed in a 12-in. case. Standard temperature and pressure assemblies are interchangeable. Controller can be modified to control two variables at one time, to include automatic reset after completion of cycle, to stop cam at any critical point in cycle, and to provide for the addition of signaling devices or the use of air or electric relays or switches. Manufacturer: Gotham Instruments Div., American Machine & Metals Inc., 149 Wooster St., New York 12, N. Y.

For additional information circle MD 2 on Page 179

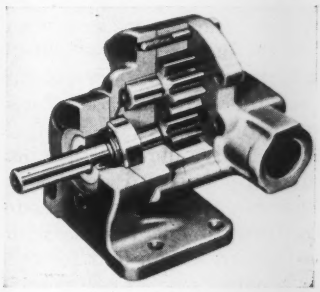
Right-Angle Gearmotors



These heavy duty NEMA - rated right-angle gearmotors are available in single-phase and three-phase models. Outputs from 1/6 to 5 hp, ratios from 5.8:1 to 100:1 and output speeds from 17.5 to 302 rpm are available. Units can be mounted in any position. Features include nickel-bronze gear, worm integral with hardened and precision-ground alloy steel shaft, semisteel gear case and Neoprene-sealed ball bearings. Drip-proof, totally-enclosed or fan-cooled types are available. Brakes are optional, and motors for special operating voltages and cycles can be supplied. Manufacturer: Abart Gear & Machine Co., 4828-36 W. Sixteenth St., Chicago 50, Ill.

For additional information circle MD 3 on Page 179

Gear Type Hydraulic Pumps

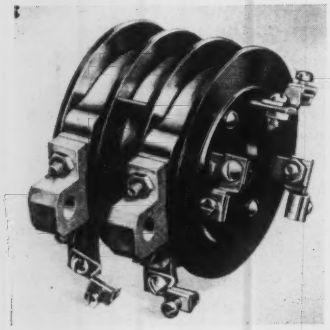


Designed for 1000 psi service with a rated capacity of from 1.5 to 46.8 gpm at 1800 rpm, these gear type hydraulic pumps are offered with either foot or flange mountings. Keying method used to attach impellers of pump to driveshaft prevents thrust loads from being transferred from shaft to impeller or end plates. Key consists of balls placed in three grooves spaced around periphery of shaft and shaftway. Impellers are thus free to float in pump cavity, despite lateral motion of shafts;

trapped fluid is utilized to pressure-lubricate shafts, bushings and journal bearings. Alloy bronze bushings, spur-type hardened steel gears and shafts and alloy iron housings are employed. Pumps are internally drained and can be assembled for either direction of rotation. Manufacturer: Adel Precision Products Corp., 10855 Van Owen St., Burbank, Calif.

For additional information circle MD 4 on Page 179

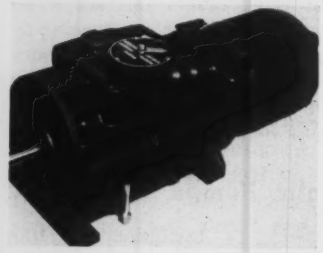
Slow-Speed Collector Ring



Furnished with one, two, three or four precision - machined conductors, this collector ring for slow-speed operation carries 75 amp at 220 v. Each conductor, designed to retain constant tension, has a double - contact full-floating alloy brush. All conductor parts are interchangeable. Measuring 5½ in. overall, the ring fits a 1½-in. tube or shaft and is easily installed. The ring body is molded phenolic. Screw binding posts are provided for quick, simple connection of the conductor cables. Manufacturer: Industrial Electric Works, Omaha, Neb.

For additional information circle MD 5 on Page 179

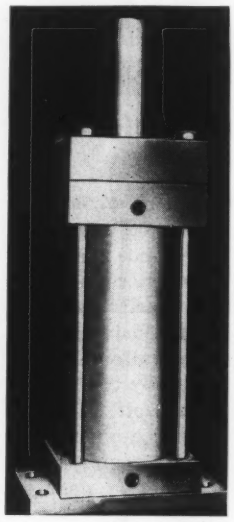
Variable-Speed Transmission



Speeds from any desired maximum to zero are provided by model 300 variable-speed transmission available in 1½ to 3-hp sizes. Lever or micrometer controls afford extreme accuracy in setting speed. Mechanical, electrical and pneumatic types of remote control can be obtained. All-metal self-lubricated unit is offered with or without motor or with built-in motors of all types. In addition to standard drive illustrated, transmission can be supplied with built-on reduction or step-up gearbox. Also available are a reversing model which delivers all speeds forward and reverse without change of input rotation and a 5-hp unit for centrifugal pump or blower applications. Transmission housing ends are machined to accommodate NEMA D flange motor, and unit can be mounted vertically for mixing or agitating applications. Manufacturer: Graham Transmissions Inc., 3754 N. Holton St., Milwaukee 12, Wis.

For additional information circle MD 6 on Page 179

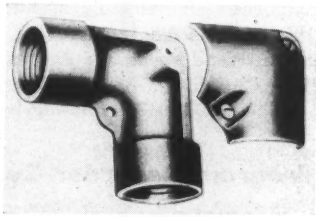
Air-Operated Accumulators



These low-pressure accumulators employ plant air at 60 to 200 psi to store fluids at up to 10,000 psi. All-metal construction adapts them for applications including severe shock loads, particularly when accumulator must completely unload itself in stopping and reversing the motion of a moving mass. Since units can be unloaded and reloaded at will, they can be used for non-pumping applications such as loading of hydraulic die cushions. By changing circuit connections, accumulators can be employed as boosters, with either plant oil or air line pressure used as input medium in developing up to 10,000 psi output pressure for operation of hydraulic work cylinders. Units are offered in variety of sizes and mounting styles. Manufacturer: Miller Motor Co., 4027 N. Kedzie Ave., Chicago 18, Ill.

For additional information circle MD 7 on Page 179

Pull-In Corner Elbow



Pull-in corner elbows, supplied in ½, ¾, and 1-in. sizes, provide a means of running a conduit around sharp corners without forcing or damaging wires. Right angle turns are possible when straddling beams or making snug corner fits. The elbow cover, which can be easily removed, is located on the side of the elbow to permit use on either inside or outside corners. Wires can be pulled in straight from either direction. Manufacturer: Gedney Electric Co., 1270 Sixth Ave., New York, N. Y.

For additional information circle MD 8 on Page 179

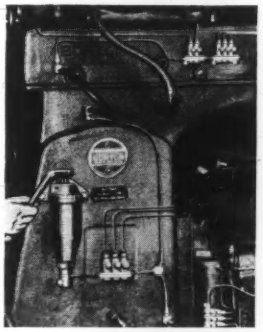
Shielded Coaxial Cable

Coaxitube consists of an inner conductor, dielectric and a seamless tube forming a coaxial pair of conductors. Because the dielectric is continuous and the outer conductor is seamless, the conductor and shield remain coaxial even when formed into intricate shapes. This design permits use for such circuit components as closely-coupled tank coils and grid feedback circuits, shielded grid leads and high-frequency transmission lines. It can be used also as a thermocouple. The dielectric can be polyethylene, Du Pont Teflon, Fiberglas or any other commercially avail-

able material to give the required electrical performance. The tube is produced in range of sizes using Copnic, Constantan or Advance inner conductor, suitable insulation and a seamless copper outer shield. Copper-Advance combination is usable up to 350 C, at which temperature it produces about 18 mv above its emf at 0 C. Outside diameter can be controlled from 0.023 to 0.100-in. or larger. Manufacturer: Precision Tube Co., 3824 Terrace St., Philadelphia 28, Pa.

For additional information circle MD 9 on Page 179

Master Oiler



This centralized lubrication system consists of a manually operated Centr-Oiler pump which supplies lubricant through a single-line circuit to SL-4 injectors. These injectors deliver a predetermined quantity of oil to each bearing every time the system is cycled. Maximum lubricant pressure is 1000 psi, and reservoir capacity is 20 oz. The system is complete with the necessary connectors and adapters for connection to existing oil holes, plain or threaded. A full range of accessories for connection to rotating or reciprocating bearings is available. Manufacturer: Lincoln Engineering Co., 5736 Natural Bridge Ave., St. Louis 20, Mo.

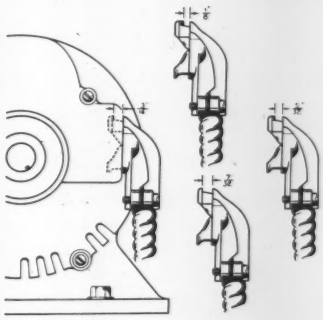
For additional information circle MD 10 on Page 179

Electronic Time Switch

By changing the external connections to the terminal board, catalog 110 electronic time switch can be used for interval timing, time delay, repeat cycling and progamming. Timing periods of 1/20-second to 4 minutes, accurate within 2 per cent, are obtained by means of a selector switch and a variable resistor. The timing capacitor is hermetically sealed with glass type terminals. The single relay employed uses a dual coil sealed against humidity. Standard units are supplied for operation on either 115 or 230 v 50/60 cycle current. All relay contacts are rated 10 amp at 115 v ac, noninductive load. Unit can be supplied in heavy-gage housing, without housing or for panel-mounting. Manufacturer: Henry G. Dietz Co., Coral Designs Div., Box 248, Forest Hills, N. Y.

For additional information circle MD 11 on Page 179

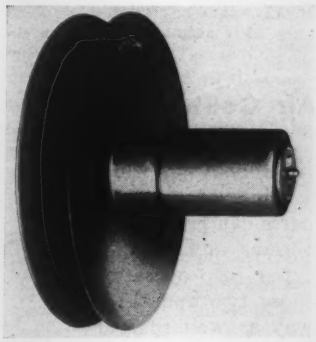
Armored Cable Connectors



This line of ⅜-in. angle type connectors for attaching armored electrical cable to ⅞-in. openings in curved motor housings is adaptable to castings of wall thicknesses of 3/64 to 7/32 in. Units are die-cast aluminum alloy made in two parts, each with a grooved lip which engages the edge and inside surface of a standard ⅞-in. opening. Operating on the cam-wedge principle, they grip tightly when two standard machine screws are tightened. Positive positioning of the cable connector assembly in any desired direction is insured, even when the unit is subjected to severe vibration. The connector is Underwriters' Laboratories approved. Manufacturer: Unimatic Corp., 52 E. Centre St., Nutley, N. J.

For additional information circle MD 12 on Page 179

Variable-Pitch Pulleys



Infinitely variable speed changes (4 to 1) in the 2 and 3 hp range are provided by Models 1050-20 and 1050-30 Roto - Cone variable-pitch pulleys. Utilizing V-to-V-belt drive, all units are dynamically and statically balanced. Rack-and - gear arrangement controls sheave movement and insures belt alignment throughout entire speed range. Pulleys are installed easily in either horizontal or vertical position without special mounting. Manufacturer: Gerbing Mfg. Corp., 650 W. Washington Blvd., Chicago 6, Ill.

For additional information circle MD 13 on Page 179

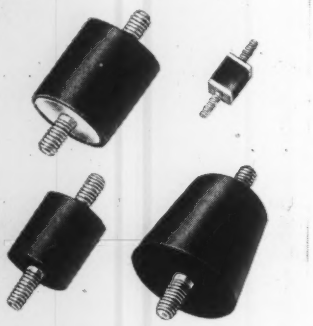
Mild Steel Electrode

High-speed production of single-pass fillet and lap welds in flat or horizontal positions is possible with Fleetweld 72 mild steel E6012 electrode. Especially suited for work where fit-up is poor, the electrode operates best at currents above those normally used with E6012 electrodes. With higher melt-off rates, greater welding speeds are attained in depositing dense, high-strength, ductile welds. Arc characteristics assure excellent root penetration, and near-flat

bead eliminates piling up of excess weld metal. Lengths of 14 and 18 in. are available in diameters of ⅛, 5/32, 3/16, 7/32, ¼ and 5/16-in. Manufacturer: Lincoln Electric Co., Cleveland 1, O.

For additional information circle MD 14 on Page 179

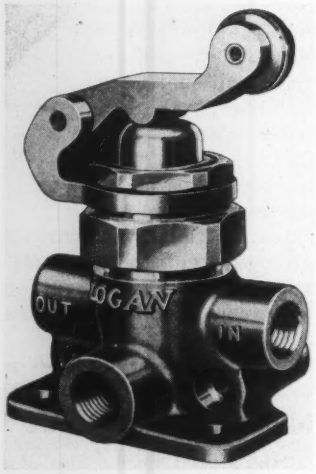
Stud Vibration Isolator



Whether installed in compression or shear, Finnflex stud-type vibration isolators effectively control transmitted vibrations. Each mount consists of a rubber cylinder rigidly bonded to steel end plates to which studs are welded. One stud can be bolted to a foundation and the other to a piece of equipment. Isolators can be obtained in 44 different sizes for load ranges from 1 to 270 lb per mount and are recommended for use with small motors, engines, lathes, and air conditioning and lighting equipment. Manufacturer: Finn & Co., 2850 Eighth Ave., New York 30, N. Y.

For additional information circle MD 15 on Page 179

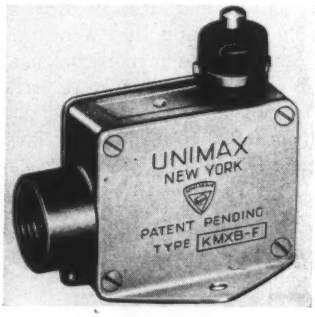
Air Control Valves



Based on single valve body to which a choice of accessory parts can be added to obtain any of five types of operation, these two and three-way as well as four-way two-position ¼-in. port air control valves employ corrosion resistant parts. The illustrated four-way two-position valve can be used as pilot operating valve for remote control of pilot-operated master valves. Two and three-way valves are made for remote control of single-acting air cylinders and are equipped with three ports, one of which can be plugged to make a two-port valve. Pushbutton, toggle, cam, air cylinder and foot operation are available. The valves can be side or base mounted. Manufacturer: Logansport Machine Co., Logansport, Ind.

For additional information circle MD 16 on Page 179

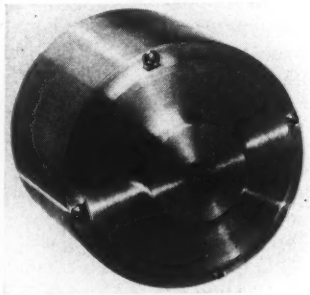
Splashproof Precision Switch



Protection against moisture, dust and splashing oil and water is insured in type KMXB snap-acting precision switch. A Neoprene boot is sealed to the actuating plunger and the entire assembly is integral with a neoprene-gasketed metal case. Seals retain flexibility at temperatures from −65 to 160 F. Basic type MX or SXX switch mounts inside KMXB Adaptacase. Unit can be obtained in styles for side mounting, base mounting from bottom and upright or inverted mounting from-the-top. Hub is tapped internally for either ½-14 or 3/4-20 conduit connection. Standard type MX basic switch is rated 15 amp 125-v ac noninductive, and type SXX basic switch, 15-amp 250-v ac. Manufacturer: W. L. Maxson Corp., 461 W. 34th St., New York 1, N. Y.

For additional information circle MD 17 on Page 179

Fractional-Horsepower Motor



Small size, light weight and efficiency rating of 70 per cent are features of this 1/30-hp motor. Built to conform to Army Air Force specification 32590, the motor operates on 115-v three - phase 400-cycle current. It has 2-in. diameter a 1⅝-in. length and weighs approximately 9 oz. Output rating up to ¼-hp can be supplied in this frame diameter. Manufacturer: Mission Electric Mfg. Co., 132 W. Colorado St., Pasadena 1, Calif.

For additional information circle MD 18 on Page 179

Phenolic Molding Compounds

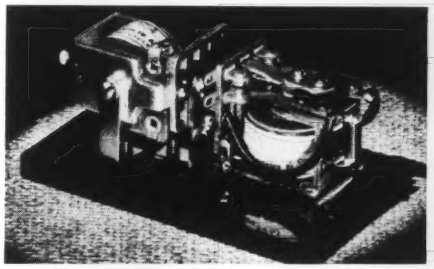
Impact type phenolic molding compounds comprise Durez 13537 natural and Durez 14482 black fabric-filled materials, and Durez 14658 nitrile rubber-bearing wood-flour-filled material. The two fabric-filled materials utilize two-step resins, have good flow properties and finish and are less critical than ordinary phenolics when electronically heated. Durez 14658 has moldability of general-purpose materials, but because of its nitrile rubber content, it possesses good impact strength and shock resistance. Low modulus of elasticity allows it to be used in thin sections

around metal inserts. All three materials can be plunger or compression molded. Manufacturer: Durez Plastics & Chemicals Inc., North Tonawanda, N. Y.

For additional information circle MD 19 on Page 179

Latch Type Relay

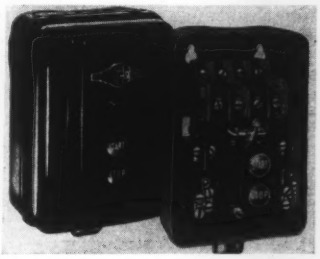
Mechanical latching relay No. 9061, for use where current is applied to coil momentarily, utilizes midget relays as basic components. When the latching coil is energized, the mechanical latch is released, allowing the armature to open. Contacts are 3/16-in. diameter, rated 8 amp 115-v ac noninductive. Arrangement of contacts is double-pole double-throw. Coils



are available for any voltage from 6 to 115-v ac, 50-60 cycles, and 2 to 32-v dc. Relay is mounted on Bakelite base measuring 1⅞ x 3⅜ x 1¾ in. Manufacturer: Leach Relay Co., 5915 Avalon Blvd., Los Angeles 3, Calif.

For additional information circle MD 20 on Page 179

A-C Magnetic Starter



In sizes to 5 hp 220-v ac and 7½ hp 440-550-v ac, this magnetic starter is furnished with dual-voltage coils, with or without built-in pilot controls and in a general-purpose or watertight enclosure. The manufacturer's Switchlet can be attached quickly to either side of the starter to provide for the addition of extra circuits. Manufacturer: Furnas Electric Co., 1045 McKee St., Batavia, Ill.

For additional information circle MD 21 on Page 179

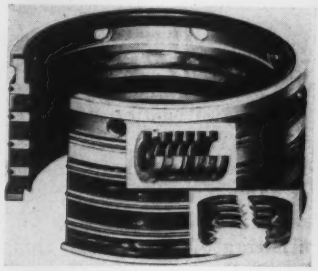
Electric Rotary Actuator

Suitable for use on either low-voltage direct current or 110-v ac, Trim Trol electric rotary actuator has been redesigned in more compact form weighing 2¼ lb. Standard gear ratio produces 350-lb-in. operating torque through 180-degree maximum rotation; static capacity exceeds 1500 lb-in. Unit incorporates zero backlash output, magnetic brake, radio noise

filter, and built-in externally-adjustable limit switches and potentiometer. Accurate positioning or electrical synchronizing controls are obtainable, and provision for mechanical synchronizing in pairs is optional. Manufacturer: Airborne Accessories Corp., 25 Montgomery St., Hillside 5, N. J.

For additional information circle MD 22 on Page 179

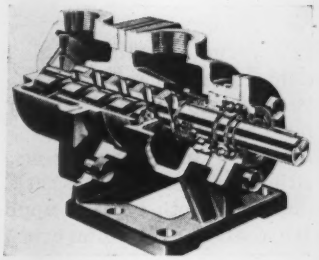
Cartridge Packing Assembly



Consisting of a specified number of lip-type seal rings separated and supported by steel thrust spacers, all permanently cemented together, this cartridge packing assembly can be obtained as integral or split type unit for reciprocating or rotary motion. It is suitable for positive sealing of fluids and soft flexible sealing lips are not subject to deformation since all pressure imposed on assembly is transmitted and absorbed by solid outer portion of cartridge. Center insert of illustration shows typical cartridge used for reciprocating motion, while lower insert depicts a split type design for use where the packing must be split to facilitate installation. Manufacturer: Emsco Derrick & Equipment Co., Cartridge Packing Div., Box 2098 Terminal Annex, Los Angeles 54, Calif.

For additional information circle MD 23 on Page 179

High-Speed Pump

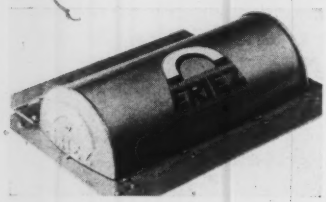


Pulsation-free IMO series A313A pump delivers to 85 gpm at up to 150 psi. It is designed for pumping petroleum products and other light or viscous fluids as required for rotary and steam atomizing oil burner, oil transfer, lubrication, hydraulic system and similar service. A power rotor and a pair of sealing or idler rotors are the only moving parts. Idler rotors mesh with

power rotor in such a manner that liquid is conveyed continuously from inlet to outlet. Pump can be direct-connected to electric motor, turbine or other high-speed drives without reduction gearing. Compact unit can be mounted in any position. Suction can be located at top, bottom, horizontal left or horizontal right. Manufacturer: De Laval Steam Turbine Co., Trenton 2, N. J.

For additional information circle MD 24 on Page 179

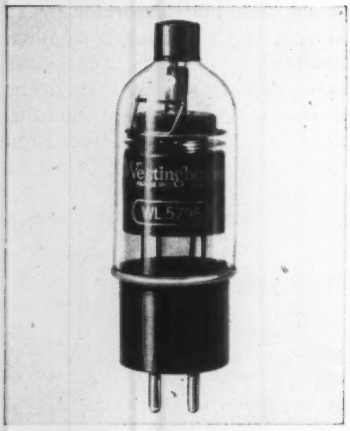
Nonelectric Plate Magnets



Supplied in three models, each with a different magnetic strength, Automagnet nonelectric Alnico plate magnets are applicable to the separation of tramp iron from materials in process. Magnet castings are fully encased by a formed cover to increase longitudinal stiffness and to prevent accumulation of iron on the backs of castings. Parts are riveted together for added structural strength. Hinges are aluminum, continuous and offset. Insulation of the plate's working surface confines magnetic strength to where it is desired. A flush air gap on the plate face prevents gathering of fine iron in this crucial area. Manufacturer: Eriez Mfg. Co., Erie, Pa.

For additional information circle MD 25 on Page 179

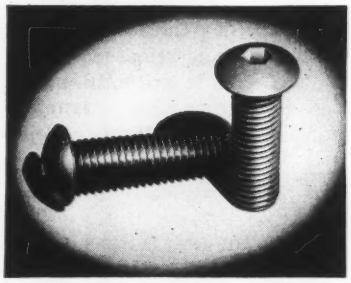
Grid-Controlled Rectifier



Type WL-5796 thyratron inert-gas rectifier is designed for industrial control and ignitor firing service. A three-electrode temperature-free tube, it attains a maximum peak anode voltage of 1500 v both forward and inverse. For general control service, maximum cathode current is 20 amp peak and 1.6 amp average. In ignitor firing applications, maximum cathode current peak is 30 amp, and average is 0.5-amp. For both types of applications, maximum negative control grid voltage is 250 v before conduction and 10 v after conduction. Maximum commutation factor rating is 10. The tube can be used in polyphase rectifiers on inductive loads with very small or no cushioning circuits. Cathode voltage is 10 v, and heating time is 10 seconds. Tube utilizes air convection cooling and can be operated in any position. Manufacturer: Westinghouse Electric Corp., Lamp Div., Bloomfield, N. J.

For additional information circle MD 26 on Page 179

Button Head Socket Capscrews



In standard sizes from No. 8 to ⅝-in. and in lengths from ½ to 2 in., these cold-forged H-K button head socket cap screws have N.C. threads held to a class 3 thread fit. The No. 10 diameter is also available with N.F. threads. Screw sockets retain true hexagonal shape, well defined corners and smooth taperless walls without deformation under repeated tightening and loosening operations. Manufacturer: Holo-Krome Screw Corp., Hartford, Conn.

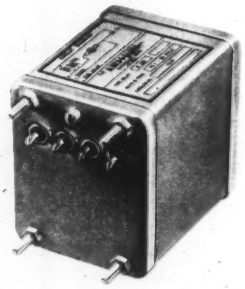
For additional information circle MD 27 on Page 179

Silver Brazing Alloy

High electrical conductivity and a remelt temperature of 1850 F are features of EutecRod 1807 silver joining alloy. Contact points and circuit breakers, tools and dies, steel assemblies which must be normalized or heat-treated, jackets and steam pipes, and other products can be joined by this alloy. Corrosion resistance is excellent, hardness is 64-70 Brinell and tensile strength is 30,000 to 32,000 psi. Sizes available are 1/64, 1/32 and 1/16-in. rods and 1/32-in. coils. Manufacturer: Eutectic Welding Alloys Corp., 40 Worth St., New York 13, N. Y.

For additional information circle MD 28 on Page 179

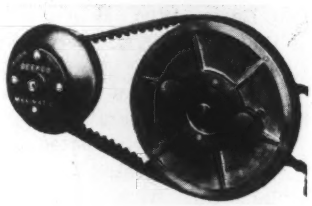
Hermetically Sealed Timers



Hermetically sealed motor timers have extended the range of the manufacturer's series of timers to include applications involving corrosive atmospheres or high humidity. All enclosures are evacuated to 100 microns and are filled with dry nitrogen so that full switch ratings can be used at high altitudes where contact capacity is normally reduced. Deterioration of motor brushes is eliminated. Manufacturer: A. W. Haydon Co., Waterbury, Conn.

For additional information circle MD 29 on Page 179

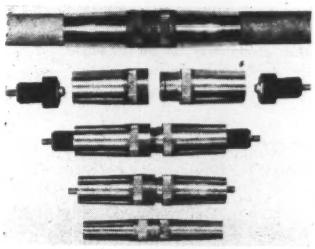
Automatic Transmission



Maximatic model AA 5-hp automatic transmission combines automatic clutch action, speed-responsive shifting, automatic torque response and infinitely varying ratios between 4:1 and 1.6:1. Performance characteristics of the unit are similar to those of a true torque converter. Principal components are high-strength aluminum-alloy pressure die castings with steel inserts for hubs. All steel parts are rustproof. The unit can be used on scooters, motorbikes, compressors and pumps. Manufacturer: Diesel Pump & Electric Mfg. Co., Culver City, Calif.

For additional information circle MD 30 on Page 179

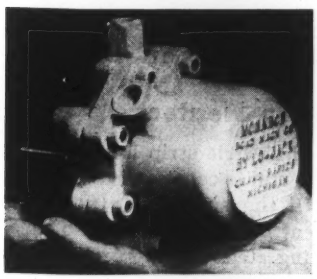
Lightweight Screw Joint



For fastening any type of solid or hollow round stock, this connecting screw joint is high-alloy aluminum tubing with machined thread and knurled exterior for convenient tightening or loosening. Sections are attached by drilling hole into the end of the rod and inserting a step-down rubber expansion piece, which expands inside hole of piece to be fastened and inside connector. This gives a slipproof fitting that will withstand tremendous pull. Screw connectors are available in stock sizes of ½, 9/16, ¾, 13/16 and ⅞-in. Other sizes and materials are obtainable on special order. Manufacturer: Minit-Joint Co., 115 S. Lafayette Blvd., South Bend 1, Ind.

For additional information circle MD 31 on Page 179

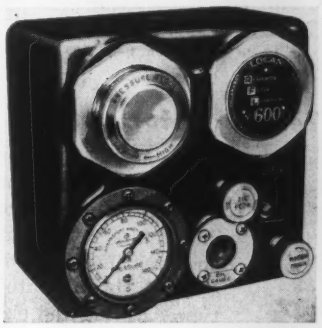
Hydraulic Pressure Pump



Intended for intermittent service, Hy-Lo-Jack oil hydraulic pressure units require less than 1 hp to deliver 1 gpm at 1000-psi pressure. Basic model is available in six types, including electric, with either foot or flange mounting. Rotation can be clockwise or counterclockwise as required. Design features include built-in relief valve and strainer as well as full needle bearing construction for balanced performance. Manufacturer: Monarch Road Machinery Co., Hydraulic Div., Grand Rapids 4, Mich.

For additional information circle MD 32 on Page 179

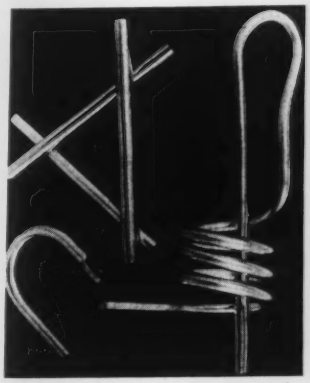
Air Regulator-Cleaner



This redesigned air regulator - filter - lubricator unit incorporates piston - type regulator, pressure gage, lubricator and synthetic-type filtering element. Filter cover and regulating valve screw into main body of unit. Measuring 6-in. sq. x 4¾ in. deep, it is installed in air intake line from main air line and can be mounted on central control panel. Two knobs on front panel provide means for accurate regulation of air pressure and amount of oil injected as required by individual job. Third knob drains water trap. Indicators provide visibility of air pressure and oil level. Manufacturer: Logansport Machine Co., Logansport, Ind.

For additional information circle MD 33 on Page 179

Chromed Steel Rods



This chromed steel rod is suitable for bending, forming, spot and butt welding, or swaging without damage to corrosion-resistant chrome finish. Nickel underplate will not chip or flake and provides corrosion - resistant base permanently bonded to steel core and chromed surface. Rods are supplied in diameters from 11 AWG to 5/16-in. in any commercial length with nickel and chromium plating to specified thicknesses. All rods are suitable for bending equivalent to limitation of unplated steel wire or rod of same diameter. Manufacturer: Kenmore Metals Corp., 380 Ninth St., Jersey City 2, N. J.

For additional information circle MD 34 on Page 179

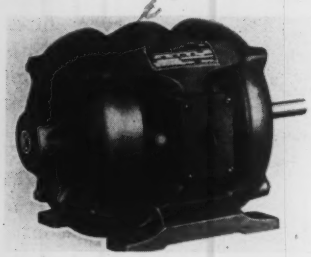
Hardfacing Alloy

Cast in rod form, Aircolite 59 is composed principally of chromium, molybdenum, carbon and iron. The hardfacing alloy is recommended for building up

surfaces subject to high stress abrasion and metal-to-metal friction where impact is moderate. Deposit maintains its high hardness at temperatures to 800 F. It has a low coefficient of friction and acquires high polish in service. Alloy is available in bare form for oxyacetylene application or coated for electric application. Manufacturer: Air Reduction Co. Inc., 60 E. 42nd St., New York 17, N. Y.

For additional information circle MD 35 on Page 179

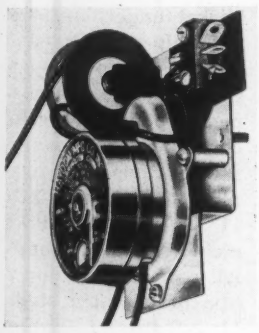
General Purpose Textile Motor



Usable where the atmosphere contains a large amount of lint, this general-purpose textile motor is a 40C open rated type. Frame and end brackets are cast iron and rotor is die cast. Mounting feet are cast integral with the frame. Oven-cured plastic insulation holds the winding in a rigid mass and completely encloses the coil end. Terminal boxes, cast integral with the frame, are closed by a steel cover with the conduit entering from any direction. Stator and rotor iron is locked and keyed in position under pressure. The motor design provides easy access to bearing housing for relubrication. Large smooth air passages permit free flow of cooling air. The motor is balanced both mechanically and electrically. Manufacturer: Century Electric Co., St. Louis 3, Mo.

For additional information circle MD 36 on Page 179

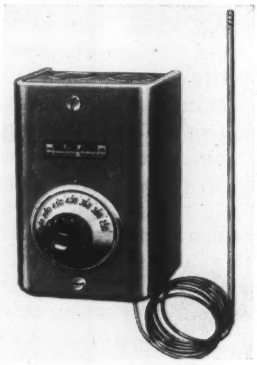
Audible Signal Timer



Available in wide range of timing intervals, series 8010 timer has buzzer signal which is actuated at preset interval and sounds continuously until manually turned off. Compact unit measures $4\frac{1}{32}$ x $2\frac{11}{32}$ x $1\frac{35}{32}$ in. It can be mounted and operated in any position, rotates clockwise or counter-clockwise and is settable in either direction, to start or when in operation. Quick-break snap-action unit is Underwriters' Laboratories rated for 16 amp 250 v ac. Basic motor operating device features 450-rpm rotor speed for quiet operation and long life. Motor is totally enclosed and has dual system of controlled lubrication. Manufacturer: Haydon Mfg. Co., Torrington, Conn.

For additional information circle MD 37 on Page 179

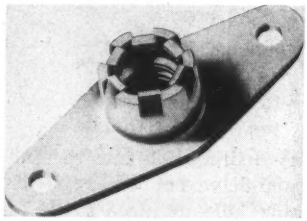
Electric Thermostat



Basic type R electric thermostat consists of an adjustable double-pole snap-action switch for various temperature settings. It is actuated by bellows connected by capillary tube to a heat-sensitive bulb. By varying amount and type of liquid in bellows, tube and bulb, any desired temperature range from 300 to 700 F can be obtained. Bulb is placed at point where temperature is to be controlled; switch mechanism and dial can be located remotely wherever convenient. Thermostat can be utilized for automatically controlling temperature of any electrically-heated liquid, solid or gas medium. In addition, it can be used as on-off switch. Manufacturer: Wilcolator Co., Elizabeth, N. J.

For additional information circle MD 38 on Page 179

Self-Locking Nut



Employing the Hex-Lok principle, Plate Lok self-locking nut, in 8/32, 10/32 and ¼-in. sizes, is essentially two nuts in one. Top locking section is formed by depressing six threaded flanges inward and downward so that they exert positive pressure against the body of the bolt without distorting or harming bolt threads. Flanges are held by tension in this out-of-phase condition and return to depressed position when bolt is removed. The all-metal unit is unaffected by oil, gas, water or corrosives. It resists all types of vibration, can be set to specific torque value and has a high reusability factor. The nut can be used at temperatures from below zero to 550 F. Manufacturer: Boots Aircraft Nut Corp., Stamford, Conn.

For additional information circle MD 39 on Page 179

Finned Electric Strip Heaters

Sheath of these finned electric strip heaters is one-piece rectangular tubing and is constructed so that all fins are in contact with the surface of the heating element to insure efficient heat transfer. Both sheath and fins are rust-resisting steel. Heater coil is nickel chromium wire insulated from inner sheath wall. Heaters operate on 115 or 230 v ac and can be used where sheath temperatures will not exceed 750 F. Fifteen standard stock sizes range from a 500-w unit

Corrosion of Bearings-2

RESISTANCE to corrosion is a factor that merits careful consideration in selecting the type of bearing to be used. This includes the period prior to installation or storage and after installation. In this treatise we will limit the discussion to the latter.

Under service conditions, temperature and the corroding medium are the controlling factors. Bearings are used in various media but these are too numerous to discuss in general terms. The usual medium is oil so it should be given first consideration.

It is difficult to know exactly what role corrosion plays under operating conditions since all that can be observed is the rate of disappearance of the bearing surface. This rate of disappearance depends on wear plus corrosion. However, from experience and laboratory corrosion tests a guide is available.

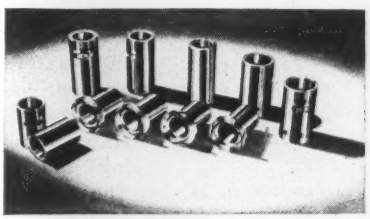
At temperatures above 200° F., and in the presence of some air, corrosion in the presence of oil may be expected. The air will oxidize the oil and acid products will be formed that are definitely corrosive to many metals. However, it is not the actual corrosion we are interested in but rather the corrosion *rate*. If the corrosion rate is one-tenth the wear rate, then it is negligible but if it is equal to or greater than the rate of disappearance due to mechanical wear then it is appreciable.

The relative corrosion of bearing metals is much different when in operation as compared to exposure such as in storage. Peculiarly, lead is again the major offender but steel is practically non-corrosive. A comparison of the tendency to corrode is approximately as shown in the chart below.

Pure lead will corrode in hot acidic oil at a very rapid rate in terms of bearing life. This rate of corrosion is substantially reduced by two factors. First, if the oil contains natural inhibitors; or if the oils are highly refined to remove the natural inhibitors and additives are used to restore this inhibition. And second, the addition of tin to lead base alloys greatly increases the corrosion resistance.

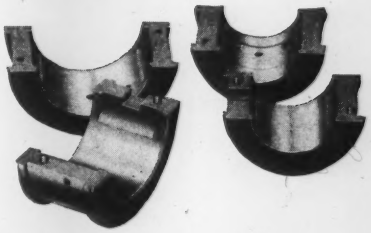
In the copper-lead alloys the rate of corrosion depends on the amount of lead in the alloy under corrosive conditions since the lead is present in free pockets. The lead cannot be protected with tin as in babbitt since the tin will alloy with the copper rather than with the lead. The only alternatives are to use copper-lead under conditions where severe corrosion is not encountered or to use an overlay of tin-base or lead base alloys over the copper-lead.

Where corrosion is a major factor in determining bearing life and yet the surface properties of a soft metal are required, a tin alloy is recommended.



Cast bronze bearings have long been a favorite with leading builders of out board motors.

If the properties of a soft metal are required and price is a major factor then lead-base babbitts may be used and corrosion resistance is obtained by the use of tin in the lead alloy. For a straight lead-tin alloy about 6 percent tin is required but in the babbitts where greater strength is required in heavy sections, the tin content may be dropped as low as 1 percent in the presence of antimony and fair corrosion resistance will be obtained.



Aluminum alloy bearings provide excellent resistance to corrosion.

For load applications intermediate to the babbitts and the bronzes, aluminum offers possibilities for excellent corrosion resistance.

Where oil temperatures run under 200° F. corrosion problems are not likely to be encountered unless water is liable to be present from some external source or due to exposure to wet conditions or condensation in outdoor environments. In this case the tin-base babbitts at low loads and the bronzes at high loads are most liable to meet requirements.

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

●

This bearing sheet data is but one of a series. You can get the complete set by writing to—

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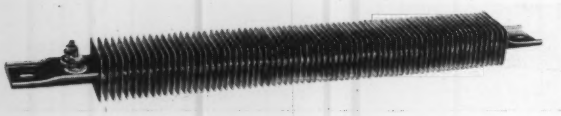
Relative Comparison of Corrosion

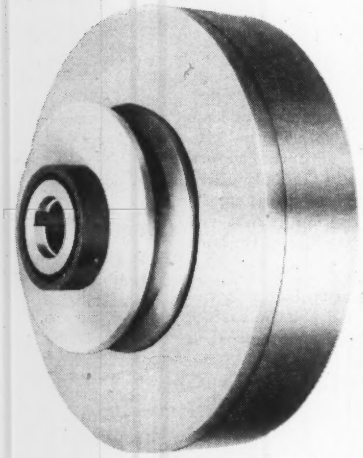
Tendency of bearing materials to corrode in the presence of hot oil and air.



with a 10½-in. overall length and a 7-in. finned length to a 3250-w unit having a 41½-in. overall length and a 39-in. finned length. Manufacturer: Vulcan Electric Co., Danvers, Mass.

For additional information circle MD 40 on Page 179

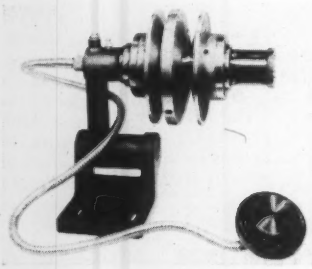
Motor Shock Absorber



Where smooth starting is required, this shock absorbing device will effectively absorb jerk occasioned in across-the-line motor starting. The unit reduces torque by permitting the motor to gain momentum before picking up entire load. There is no power loss and no slipping. Device can be furnished for either forward or reversing drives in sizes for ¼ to 3-hp motors. Manufacturer: Crown Mfg. Co., 1025 N. Sixth St., St. Louis 1, Mo.

For additional information circle MD 41 on Page 179

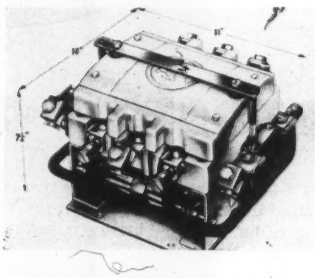
Variable-Speed Pulley



Stepless speed control is obtained with Adjust-O-Matic double-groove single-belt pulleys. Plastic models equipped with steel-faced sheaves and designed for drives of ¾ to 1 hp provide speed ranges of 3:1; aluminum alloy model delivers 2:1 range on drives of 1, 1½ and 2 hp. Positive sheave location minimizes ratio slip under varying or cyclic loads. Set by mechanical control, spacing between sheaves is held regardless of belt wear, pull or stretch. Unit can be mounted in any position without causing undue belt wear. Knob control, indicator dial or remote control can be supplied. Manufacturer: American Type Founders Inc., 200 Elmora Ave., Elizabeth B, N. J.

For additional information circle MD 42 on Page 179

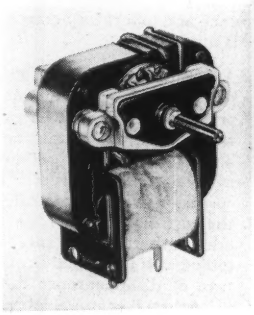
Magnetic Starter



Compactness is achieved in this size 4 RA magnetic starter by use of a right-angle balanced mechanism. A knee-action bellcrank fulcrum multiplies leverage and transfers it from vertical to horizontal plane. Four-pillar guided alignment insures straight line action and positive contacting. Wiring is straight through, and design of starter permits ample working space for ease of installation and servicing. Terminals are accessible from the top; load and line wires are connected at opposite ends. Manufacturer: Arrow-Hart & Hegeman Electric Co., 103 Hawthorn St., Hartford 6, Conn.

For additional information circle MD 43 on Page 179

Fractional-Horsepower Motor



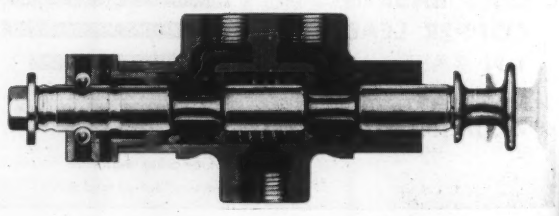
Suitable for use in fans, blowers, heating equipment, etc., type 230 fractional-horsepower motor has a flat speed-torque curve and high starting and pull-up torque. A uniform air gap around the stator results in even cooling and quiet operation. The motor is produced in two models with ratings from 1/200 to 1/50-hp. Manufacturer: Russell Electric Co., Chicago, Ill.

For additional information circle MD 44 on Page 179

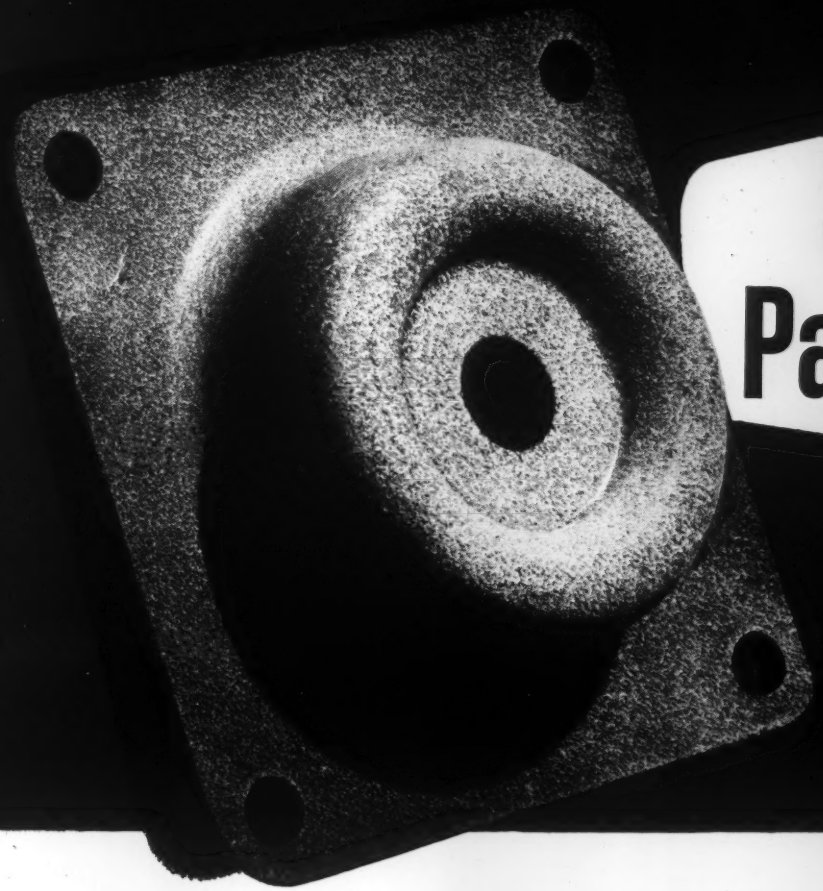
Neutral Air Valve

When this air valve is in the neutral position, all ports are blocked. Consequently, the piston of a single or double-acting cylinder can be held in any position. Ranging in pressure from zero to 175 psi the valve can be supplied with knob, lever, clevis or treadle type operating means. Pipe sizes of ¼, ⅜, ½, ¾ and 1-in. are available. Manufacturer: Valvair Corp., 454 Morgan Ave., Akron 11, O.

For additional information circle MD 45 on Page 179



FOR TIGHT SEALING OF UNIVERSAL ACTION



Sirvis Packings

This is a custom-built fulcrum seal. Designed specifically for tractor use, it seals the opening through which the gear shift lever passes into the transmission. This Sirvis leather prevents all foreign matter from entering the transmission, and even under toughest operating conditions, it makes certain dependable performance.

Sirvis packings, like all Sirvis mechanical leathers, are made to deliver long, rugged service. Engineered from the unblemished center portion of select steerhides, they are used to seal oil, water, air or other fluids and gases under various pressures and temperatures.

When you need positive protection for vital machine parts, consult Chicago Rawhide Sirvis engineers. Drawing upon their vast knowledge, experience and manufacturing facilities, they will supply a Sirvis boot, cover, diaphragm or packing to fulfill your most exacting requirements.

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SIRVIS

ENGINEERING DEPARTMENT
EQUIPMENT

For additional information on this new equipment see Page 179

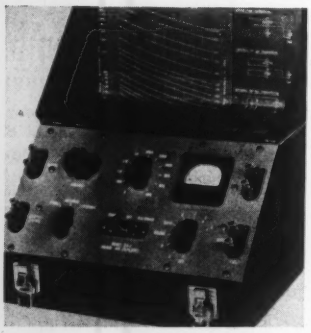
Dry Film Thickness Measure

Thickness of dry films and thin sheet metals, foils, paper or plastic can be determined quickly by method using Interchemical direct reading thickness gage designed originally for wet film measurement. Attached dry films are measured visually by means of optical attachment to gage plus suitable light source. Film is stripped at two points ½-in. apart, and two outside gage wheels rest on these areas. Gage is rotated until its eccentric center wheel touches film. Light source permits observer to check point at which center wheel meets film and cuts off light. Direction of gage rotation can be reversed and average of two readings used for greater accuracy.

For measuring thin metal sheets, foils, paper and plastic films, feeler method is used. Gage rests on plane surface with center wheel at maximum reading. Narrow strip of sheet material is placed on plane surface between two outer wheels, and gage is rotated over strip until slight binding is noticed as strip is slid under measuring surface. Reading at that point gives thickness. Manufacturer: Henry A. Gardner Laboratory Inc., Bethesda 14, Md.

For additional information circle MD 46 on Page 179

Dynamic Strain Amplifier



Usable with any standard cathode ray oscilloscope, model BA-1 portable strain amplifier measures almost any mechanical reaction detected by SR-4 bonded resistance wire strain gages. Instrument can be used for load measurement; stress analysis and function analysis during operation of equipment to show amplitude, frequency and wave form of sharp transients; steady state oscillations; and static loads or their combinations. Displacement, acceleration, vibration, damping, timing, weight, and pressure can be seen and measured during operation of machines or mechanical devices. Data can be recorded by means of either special high-speed camera or still camera without shutter.

Instrument is complete package of bridge elements, signal chopper, calibration system, amplifier and power supply. Internal bridge elements handle either single-gage dynamic measurements or two-gage static and dynamic measurements. Switch clears circuit for use of entirely external bridge system connecting into amplifier input. Manufacturer: Baldwin Locomotive Works, Philadelphia 42, Pa.

For additional information circle MD 47 on Page 179

Industrial Television System

This television system facilitates remote viewing of any action at distances up to 500 ft. It utilizes small 8-lb camera incorporating only three tubes, including Vidicon pick-up tube, and suitcase-size control box combining power supply, synchronous generator, amplifying and deflection equipment and 7-in. kinescope monitor. Entire system operates on 110-v 60-cycle ac and consumes 375 w. Signals are transmitted from camera to viewer over coaxial-cable closed circuit.

Progress of multiple long-time tests at different places can be followed from one central point. Camera facilitates inspection of underside or interior of products and materials. Other possibilities include comparison of television image with standard either optically or electrically and use of system to provide signals to motivate automatic controls or indicators. Still photography can make permanent records from screen of monitor, while motion picture photography can be used to record high-speed data for later examination. Ability of television to detect infrared and ultraviolet radiations offers further possibilities for industrial applications. Manufacturer: Radio Corp. of America, RCA Victor Div., Scientific Instruments Section, Camden, N. J.

For additional information circle MD 48 on Page 179

Practical tolerances for resilient gaskets

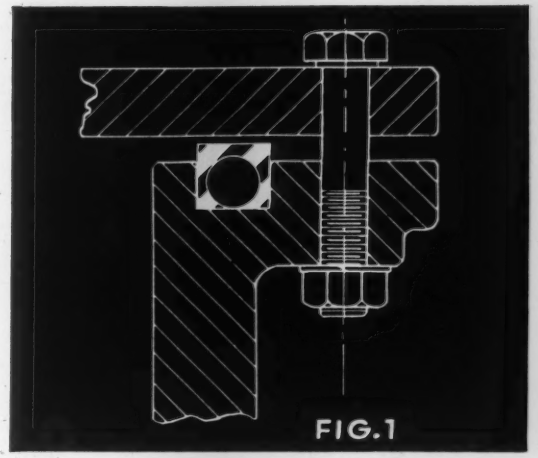


FIG. 1

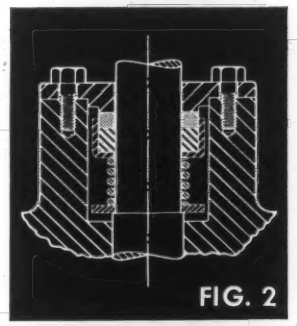


FIG. 2

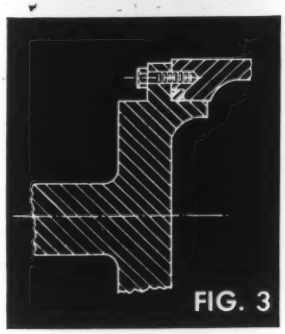


FIG. 3

Close tolerances in resilient gaskets are rarely necessary. During assembly, gaskets are usually compressed to a predetermined thickness or percentage compression. Tolerances do not become critical if the gasket material chosen seals effectively over a wide range of compressions.

Typical of such wide range materials are Armstrong's Cork-and-Rubber Compositions. Their recommended compression range of 20% to 33% largely eliminates the need for gasket tolerances closer than ± .010".

On confined gasket applications, too, close tolerances are unnecessary with cork-and-rubber. Because these materials are truly compressible, they actually decrease in volume under load. This enables gaskets made slightly oversize to compress into a completely confined space without provision for lateral flow.

Figure 1, for example, illustrates the relative size of the cork-and-rubber gasket and the comparable non-compressible molded ring. Wide tolerances obviously would have much less effect on the greater thickness of the cork-and-rubber composition.

Likewise in figure 2, wide tolerance cork-and-rubber gaskets make an effective packing ring. Their true compressibility controls lateral flow and prevents excessive build-up of radial pressure on the shaft.

Non-compressible rings on the flanged journal of a calender roll in figure 3 required extremely close tolerances. Undersized rings leaked. Oversized rings prevented metal-to-metal contact between the flanges and caused misalignment. Both of these problems were solved with standard tolerance cork-and-rubber rings.

Wide tolerances are but one type of saving made possible by Armstrong's Cork-and-Rubber. We suggest that you call your Armstrong representative. He can help put these versatile materials to work on your application.

Send for this Gasket Handbook

You'll find useful application and specification data in the revised 24-page booklet, "Armstrong's Gasket and Sealing Materials." It contains up-to-date information on straight synthetic rubber, cork-and-synthetic-rubber, and cork composition gasket and sealing materials.

This booklet includes ten technical discussions of the factors influencing modern gasket and joint design. It also suggests methods of putting Armstrong's stock materials to specialized uses in such fields as radio, electrical, automotive, petroleum, and transportation industries.

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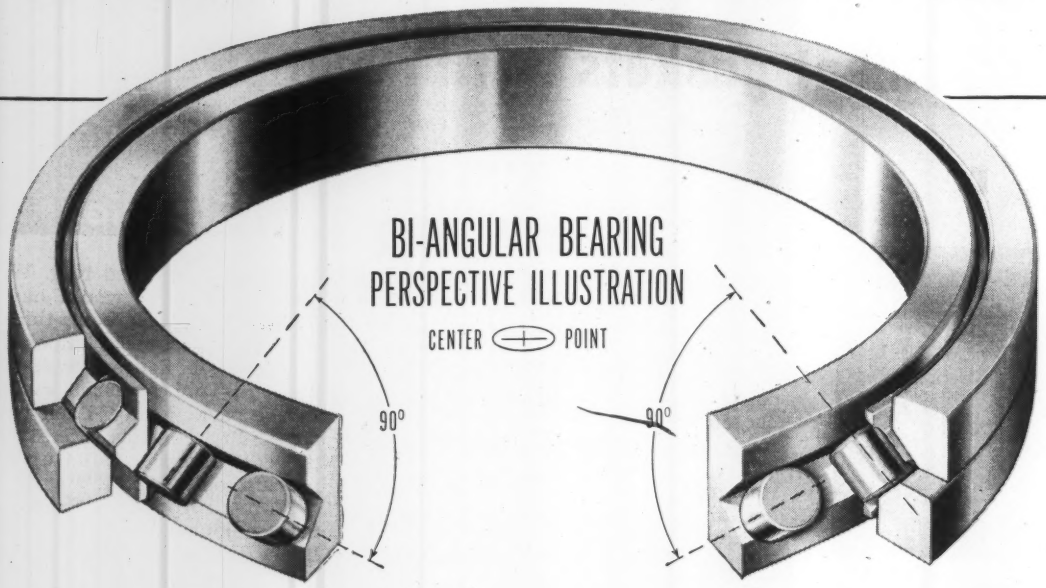
KAYDON BI-ANGULAR ROLLER BEARINGS



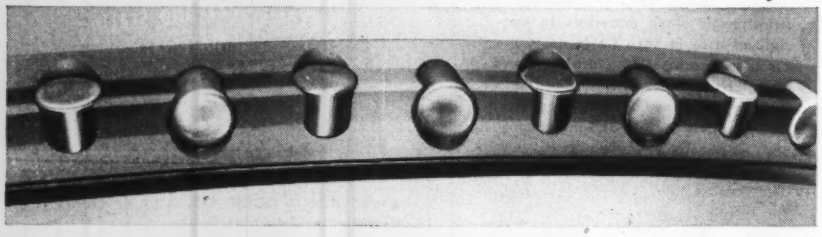
Does your bearing problem involve *both* thrust *and* radial loads? **KAYDON** BI-ANGULAR Roller Bearings are adaptable to various proportions of thrust and radial loads, as indicated below. These bearings are particularly suitable for low speed applications involving impact loads.

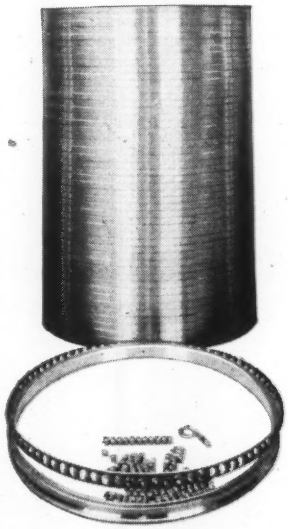
Whatever *your* bearing needs may be, **KAYDON** has *all* the facilities your engineers require. Whether you need only a few or many special bearings 4″ to 120″ outside diameter, or millions of high precision rollers, contact **KAYDON** for confidential counsel and cooperation.

ADAPTABLE TO EQUAL OR UNEQUAL THRUST AND RADIAL LOADS

For thrust loads *equal* from both directions, **KAYDON** BI-ANGULARS are made with every other roller reversed, as shown here. Adjacent rollers are at 90° angles to each other, permitting this bearing to take either radial loads or thrust loads, or a combination of both.

To handle heavy thrust load *greater* from one than the other direction, the bearing can be made with every second, third or fourth roller reversed, depending upon how relatively *unequal* the loads may be. Write for further detailed information.





31.000″ x 34.988″ x 2.000″ **KAYDON** BI-ANGULAR Roller Bearings have been produced in large quantities and are successfully in use. They are further proof of **KAYDON** ability to design and make unusually large, precision bearings for specific needs.

KAYDON Types of Standard or Special Bearings: Spherical Roller • Taper Roller Ball Radial • Ball Thrust • Roller Radial • Roller Thrust • BI-ANGULAR Roller

THE KAYDON ENGINEERING CORP., MUSKEGON, MICH.

• ALL TYPES OF BALL AND ROLLER BEARINGS 4″ BORE TO 120″ OUTSIDE DIAMETER •

HELPFUL LITERATURE

FOR DESIGN EXECUTIVES

71. Hydraulic Cylinders

Lesoon Mfg. Co.—2-page illustrated bulletin CS-849 describes application of hydraulic cylinders in concrete block machine. Details of operation of heavy-duty cylinders are discussed.

72. Reproduction Machine

Ozalid Div., General Aniline & Film Corp.—6-page illustrated folder depicts Printmaster photographic reproduction machine designed for handling wide range of work on large volume basis. Suggestions are given relevant to using machine in conjunction with other models or set up as all-purpose unit.

73. Electric-Air Valve

Bellows Co.—6-page illustrated folder No. AU-350 explains design, operation and application of Electroaire valve. This packaged air control unit consists of a four-way directional slide valve, solenoid controls and piston speed regulators.

74. Plastics

E. I. du Pont de Nemours & Co., Polychemicals Dept.—12-page illustrated bulletin "Du Pont Plastics" outlines properties, available forms, working techniques and uses of polythene, nylon, Teflon and Lucite. Brief descriptions of Butacite, Plastacele, Pyralin and Rulan plastic materials are included.

75. Small Motors & Blowers

Heinze Electric Co.—16-page illustrated brochure "Sub-Fractional Horsepower Motors" presents basic line of motors and blowers for continuous or intermittent service, for horizontal or vertical operation and with variety of variations or accessories.

76. Silicon Laminations

Thomas & Skinner Steel Products Co.—Illustrated bulletin deals with oriented silicon laminations which permit making compact transformer stacks while maintaining high permeability with low core loss.

77. Coolant Pumps

Pioneer Pump & Mfg. Co.—8-page illustrated bulletin "Pumps" depicts line of coolant and lubricant pumps and shows numerous applications of these units on various industrial machines.

78. Fabricated Aluminum

Revere Copper & Brass Inc.—20-page illustrated booklet "Revere Aluminum Products" contains descriptive information about various forms in which aluminum is fabricated. Illustrations of products employing aluminum alloys in form of extruded shapes, tube and coiled sheet are included.

79. Rotary Switches

Electro Switch Corp. — 12-page illustrated catalog No. 1956-P on Esco type P rotary switches gives detailed information on use of these multipole snap switches for wide variety of services in control and transfer of electric circuits.

80. Fabricating Calculator

Wales-Strippit Corp.—Circular time-estimating slide rule aids in computing time and cost of each part and complete run on Wales metal fabricator while in planning stage.

81. Automatic Valves

A. W. Cash Valve Mfg. Corp.—8-page illustrated bulletin No. 283 lists changes, additions and improvements in pressure reducing and regulating valves and also introduces automatic reseating relief valve that affords temperature and pressure relief for both domestic and industrial type water heaters.

82. Reproduction Method

Haloid Co.—8-page illustrated form No. X-100 depicts Xerography dry electrical process of printing, picture taking, reproducing documents and drawings and making offset master paper plates for duplicating and printing. Sensitized paper is unnecessary since prints can be made on ordinary papers, cloth, glass and ceramics; colored prints can be made by using colored developing powders.

83. Wire Forms & Assemblies

H. H. Titchener & Co.—8-page illustrated folder on wire forms, wire and strip steel assemblies, light stampings, welded wire assemblies, guards, grills and shelves shows typical parts which can be produced to specification.

84. Cellular Rubber

Sponge Rubber Products Co.—12-page illustrated "Sponge Cellular Rubber" gives engineering and application data on cellular rubber which affords airtight moisture-proof insulation against passage of dirt, liquids, noise and vibration.

85. Rotary Pumps

Blackmer Pump Co.—6-page illustrated bulletin No. 307 covers construction, maintenance, operation and application of rotary pumps. Details of sliding and swinging vanes are given along with specifications and other useful data.

86. Tubular & Split Rivets

Chicago Rivet & Machine Co.—12-page illustrated catalog No. 39 contains complete list of various types of rivets including those made of brass, steel, copper, aluminum, nickel silver, stainless steel and Monel.

87. Name Plate Stampings

Dayton Rogers Mfg. Co.—4-page illustrated mailing piece describes how service which supplies die-cut metal stampings can save money for users of relatively small numbers of name plates.

88. Squirrel-Cage Motors

Star-Kimble Electric Co.—4-page illustrated bulletin B-201 contains construction and assembly details of polyphase squirrel-cage motors. Dripproof units are available in three different types in sizes from ½ to 600 hp.

89. Air-Hydraulic Coupler

Snap-Tite, Inc.—4-page illustrated Vol. 1, No. 4 of "Snap-Tite Bulletin" presents 4-in. automatic coupler designed for air, water and other low viscosity fluids.

90. Liquid Level Control

Taylor Instrument Cos.—6-page illustrated bulletin 99069 describes Taylor force-balance liquid level transmitter which is designed to measure pressure and liquid level of viscous or corrosive liquids and liquids containing solids.

91. Industrial Photography

Eastman Kodak Co., Photographic Div.—16-page illustrated booklet "Functional Photography in Industry" sums up, in nontechnical language, processes and techniques employing photography to research, quality control, production, training, advertising and sales. Discussed are projection equipment, document copying machines, microfilming, high speed motion pictures, photomicrography and analysis by light.

92. Cast Iron Pulleys

Sprout, Waldron & Co.—16-page illustrated bulletin 25 and supplementary bulletin 25 are devoted to complete line of Elite-Face pulleys. Specifications and prices are given on pulleys in diameters ranging from 4 to 40 in. Belt life increases of as much as 400 per cent are obtained through use of Belt-Saver tail pulley on bucket elevators and belt conveyors.

FOR MORE INFORMATION

on developments in "New Parts" and "Engineering Department" sections—or if "Helpful Literature" is desired—circle corresponding numbers on either card below

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93. Extruded Aluminum Shapes

Aluminum Co. of America—12-page illustrated bulletin "Alcoa Aluminum Extruded Shapes" covers design and production advantages, gives history examples, design and cost considerations, terminology and data on this form of material.

94. Reproduction Machines

Paragon-Revolute Corp.—24-page illustrated catalog "Revolute Print-Making and Processing Equipment" discusses various types of print-making processes and their advantages. Described are automatic blueprinting machine, Dimo type printers and finishers, printing machines, print dryers, washing and developing trays and other equipment.

95. High Pressure Pumps

Denison Engineering Co.—4-page illustrated bulletin PU-4 describes high pressure pumping units designed to meet recommendations of JIC Hydraulic Standards for industrial equipment. Pumps can be obtained with either constant or variable volume to deliver from 9 to 32 gpm at pressure up to 5000 psi.

96. Locks & Hardware

American Hardware Corp., P. & F. Corbin Div.—60-page illustrated catalog K294 for use by industrial engineers and designers covers complete line of locking, catching and swinging devices. Dimensions, engineering drawings and construction details of each device are included. Shown are pin tumbler key-operated devices, utility locks, handles, hinges, catches and other hardware items.

97. Plug-In Construction Components

Alden Products Co. — 12-page illustrated loose-leaf booklet "Na-ald Components for Plug-In, Unit Construction" describes electronic equipment components which must go together easily, stay in place, yet come apart quickly when repair or replacement is necessary. Slide-in lock-in chassis, back connectors, plug-in bases and captive screws are some of the units on which design details are given.

98. Centralized Lubrication

Farval Corp. — 8-page illustrated bulletin "Studies in Centralized Lubrication" cites case studies showing savings effected through installation of centralized lubrication systems on hydraulic turbo-generator, rock crushers, rolling mill, forging press and printing press.

99. Electrical Enclosures

Kirk & Blum Mfg. Co.—16-page illustrated booklet 838 describes and shows control desks, cubicles, electrical cabinets, power distribution and control panels, switch gear housings and outdoor electrical enclosures.

100. Nut Retainer

Tinnerman Products, Inc.—8-page illustrated bulletin No. 245-S covers 49 stock sizes of Speed Grip nut retainer. Specifications of each stock size, screw thread size, panel range and other dimensions are listed.

101. Electric Timing Motors

Haydon Mfg. Co.—8-page illustrated catalog No. 322 gives comprehensive listing of electric timing motors complete with dimensions, circuit diagrams, data on standard specifications and ratings of each unit.

102. Fluid Pressure Cells

Baldwin Locomotive Works, Testing Equipment Div.—2-page bulletin 300 pictures and gives complete specifications of SR-4 resistance wire strain gage type pressure cells in eight capacities from 300 to 50,000 psi.

103. Safety Switches

Trumbull Electric Mfg. Co.—8-page folder TEC-10 gives information on high capacity interrupter safety switches for use in industrial distribution systems. Grid pin design breaks up arc and interrupts heavy loads quickly and cleanly.

104. Micro Hardness Tester

Kent Cliff Laboratories—4-page illustrated bulletin K-49 portrays Kentron micro hardness tester which applies dead weight loads from 1 to 1000 grams and can be used in conjunction with both Knoop and 136° diamond pyramid indenter.

105. Centrifugal Pumps

Eastern Industries, Inc.—12-page illustrated bulletin 100 describes over 100 close-coupled models of centrifugal pumps in sizes from 0 to 70 psi and 0 to 70 gpm.

106. Automatic Clutches

Automatic Steel Products Inc., Mavury Clutch Div.—6-page illustrated bulletin 216 outlines features of automatic torque control unit for gasoline engines and electric motors which employs mercury as control medium.

107. External Hone

Sunnen Products Co.—4-page illustrated bulletin X-MAN-5003 describes external hone recommended for all pins, spindles, plungers, shafts and other parts with diameters from 0.120 to 2.750-in. Accuracy of work is held within 0.0001-in.

108. Hydraulic Accumulators

Greer Hydraulics, Inc.—4-page illustrated bulletin 409 explains use of Hydro-Pneumatic accumulator as means for suppressing line shock and provides data for determining size of accumulator required.

109. Check Valves

Durable Mfg. Co.—6-page illustrated folder form C.V.1-50 presents details of line of check valves which can be installed in all standard pipe fittings. Having only three valve parts plus seat and locking ring for easy assembly, unit operates in any position on any liquid, gas or air service.

110. Magnetic Controls

Furnas Electric Co.—14-page illustrated catalog No. 50 covers various magnetic controls for industrial applications including starters, contactors, relays and related accessories.

111. Drafting Templates

Rapidesign, Inc.—8-page illustrated catalog No. 50 depicts and describes 30 types of templates for use by draftsmen, engineers, architects and designers. Typical made-to-order templates are shown, and prices of all types are listed.

112. Copper & Copper Alloys

American Brass Co.—26-page illustrated publication B-34 is specification index for copper alloys. Applicable specifications such as grade, type and temper as well as general characteristics of most-used alloys are covered.

113. Protective Coatings

Houghton Laboratories—4-page bulletin covers Hysol industrial coatings for corrosion prevention that include Corrodon and Corsdal types that can be used to improve present paint systems by serving as undercoat or as system based on use of Corsdal over Corrodon as undercoat to achieve high adhesion, toughness and protection.

114. Flexible Shaft Units

Elliott Mfg. Co.—4-page illustrated circular 199 gives engineering fundamentals of heavy duty flexible shaft units, data regarding torque and horsepower, and information on 26 different sizes and types of units which are available from stock.

115. Regulating Valve

M-B Products—4-page illustrated bulletin No. 1000 is concerned with automatic pressure reducing and regulating valve that can be set to desired pressure without bleeding air line. Component parts and cross-section view of valve are shown also.

116. Laboratory Microscopes

Bausch & Lomb Optical Co.—24-page illustrated catalog D-125 describes complete line of laboratory type microscopes and their accessories. Also covered are accessories such as integral illuminators, various types of eyepieces, condensers and mechanical stages.

117. Molybdenum Fabrication

Fansteel Metallurgical Corp.—8-page bulletin No. 4.303 presents methods and procedures for forming and fabricating molybdenum and contains instructions for forming, punching, shearing, drawing, spinning, etching and hydrogen firing.

118. Blind Rivets

B. F. Goodrich Co. — 20-page illustrated "Rivnut" data book for designers and engineers explains installation, typical applications and advantages of using these threaded one-piece blind fastenings which also serve as nut plates. Detailed are types for use with various materials, strength requirements, types of heads, torque, seal and grip as well as tools for applying Rivnuts.

119. Solenoid Valves

Skinner Electric Valve Div., Skinner Chuck Co.—6-page illustrated bulletins Nos. 501 and 505 are descriptive of stainless steel solenoid valves and stock solenoid valves, respectively. Complete engineering, size and operating data are included.

120. Motors & Generators

Janette Mfg. Co.—4-page illustrated bulletin No. 15-1 lists motors and generators capable of supplying from 6 to 250-v dc or 115 and 230-v ac at frequencies of 60 to 50, 50 to 60, 60 to 25, and 25 to 60 cycles. Power range extends from 50 to 6000 w dc and from 110 to 4500 va at 50 to 100 per cent power factor.

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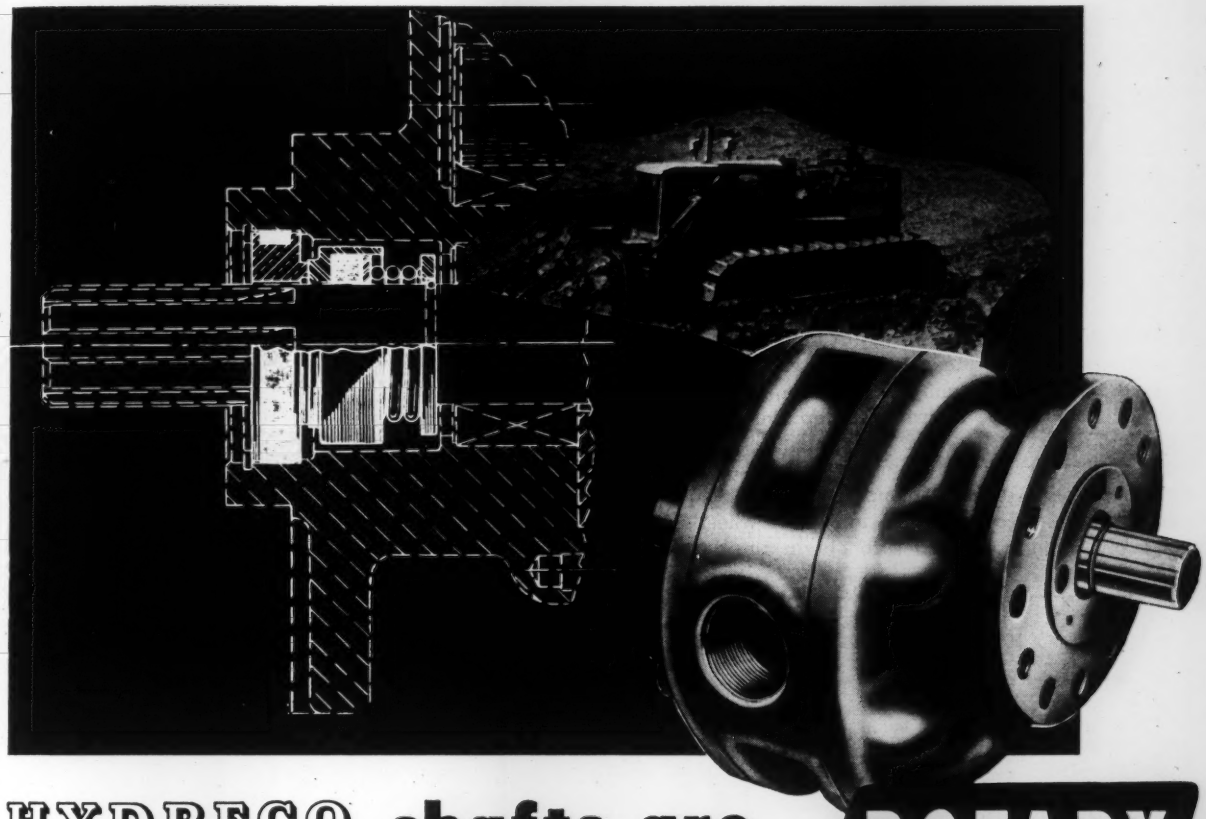

HYDRECO shafts are *Sealed for Certain* with ROTARY SEALS

"Designed to take abuse"—that's the slogan Hydraulic Equipment Co. of Cleveland uses to describe their gear pumps. They're built to give trouble-free performance, and keep on giving it, in such rugged applications as providing the hydraulic power for the dozer blade on the heavy duty Baker Bulldozer illustrated. In equipment like this, "maybe's" won't do for vital components. That's why you'll find Rotary Seals at the heart of Hydreco units—to provide *Certainty* in the critical factor of Shaft Sealing.

This is a special adaptation of the basic Rotary Seal principle—one of the many developed for manufacturers large and small to meet the requirements of specific demands in widely varying fields. If *you* have a tough Sealing problem, Rotary Seal engineers can help you answer it—and, if called in at the drawing-board stage, can often suggest the simplest design approach from a Shaft Sealing standpoint, based on broad and successful experience in many lines. Write us today!

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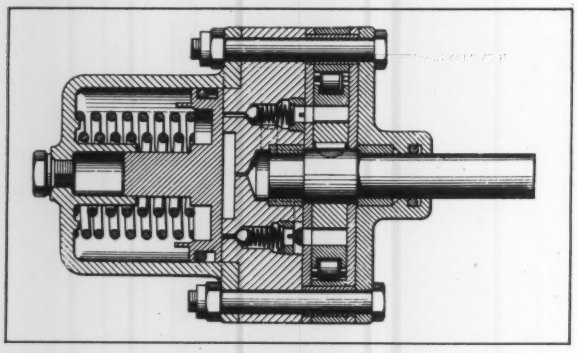
THE ROTARY SEAL COMPANY

mechanical seals for rotating shafts

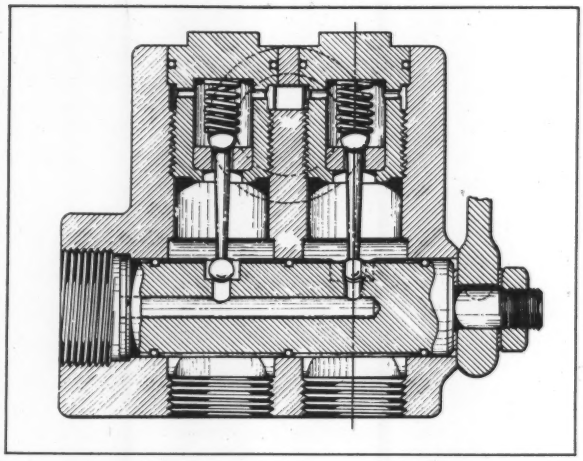
NOTEWORTHY PATENTS

ANGULAR DAMPING of objectionable oscillation in members such as steering or castering wheels is provided by an antishimmy device comprised, essentially, of a gear type hydraulic pump and a variable orifice. The inner gear of the pump is mounted on the shaft of the wheel or caster to be damped and the outer gear of the pump is secured to the frame or body. Rotary oscillation of the wheel shaft pumps oil through an orifice to obtain the damping



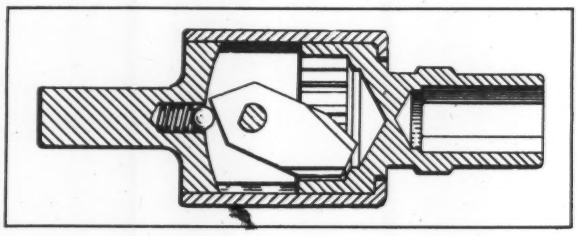
effect. A spring-loaded piston serves to keep the hydraulic system under pressure and also acts as a safety relief valve for the pump. Connected to the orifice is a bimetallic strip which automatically varies the size of the orifice opening with temperature changes to give constant damping operation independent of temperature conditions. Bell Aircraft Corp. has been assigned the patent, No. 2,502,523, by Arthur S. Irwin.

ELIMINATION OF CAMS for actuating the stems of poppet type fluid directional-control valves is accomplished by substituting ball seats on the rotary control shaft. Such an arrangement is covered in patent 2,494,753, assigned to Vickers Inc. by Duncan B. Gardiner. Fluid pressure is introduced to the valve through two passages which would normally be held closed by spring-loaded balls. These balls can be lifted from their seats by push rods or stems operating from seats on the control shaft. The control shaft end of the stems also mount balls which act in valve seats on the shaft to seal the two exhaust passages. Valve stem seats are so located, angularly, that only one of the upper valves and the other lower valve can be open at a particular angular setting of the control shaft. With the control shaft rotated all



the way to the right, fluid under pressure flows past the upper ball valve, around the control shaft and to the pressure side of the motor, piston or other hydraulic device being operated. With the control valve in its right position, the other valve stem has lifted its lower ball from its seat, opening a passage from the exhaust side of the piston or other fluid operated device. Turning the control shaft in the opposite direction reverses flow in the hydraulic lines.

REVERSIBLE RATCHET mechanism disclosed in patent 2,505,678 and assigned to Standard Pressed Steel Co. by Francis R. Kull utilizes a simple spring-loaded, toothed pawl and requires no more space than a standard socket wrench head. Teeth on the pawl, which is pinned on the output shaft of the mechanism, engage notches in the circular, hollow head of the input shaft. Pawl teeth and driving notches are so located and shaped that increasing the input torque increases the locking action between tooth and notches. Reversing the applied torque causes the tooth and notches to disengage, with ratcheting ob-



Genuine ALEMITE LUBRICATION FITTINGS

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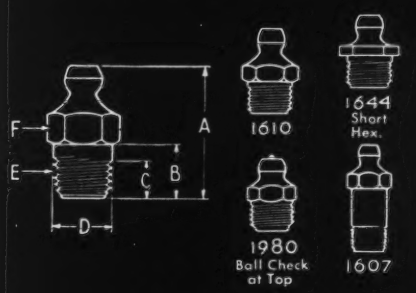
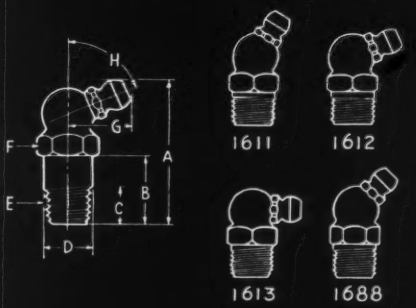


STRAIGHT HYDRAULIC FITTINGS

Part No.	A	B	C	D	E†	F
1607	1 17/64″	25/32″	13/64″	13/32″	1/8″ P.T.	7/16″
1610	37/64″	17/64″	13/64″	25/64″	1/8″ P.T.	7/16″
1618	1 7/64″	5/16″	1/4″	1/2″	1/8″ P.T. (Female)	1/2″
1634	13/16″	5/16″	1/4″	13/32″	1/8″ P.T.	7/16″
1644	21/32″	5/16″	1/4″	13/32″	1/8″ P.T.	7/16″
*1650	57/64″	3/8″	19/64″	13/32″	1/8″ N.P.T.F.	7/16″
1669	1 3/64″	1 17/64″	1/4″	13/32″	1/8″ P.T.	7/16″
1684	2 5/8″	2 3/16″	5/16″	13/32″	1/8″ P.T.	3/8″
*1708	53/64″	23/64″	19/64″	13/32″	1/8″ N.P.T.F.	11/32″
1980	41/64″	17/64″	7/32″	13/32″	1/8″ P.T.	7/16″

*Leak-proof seal
†Thread size listed as 1/8″ P.T. is 1/8″ P.T.F.—SAE EXTRA SHORT



1/8″ THREAD ANGLE HYDRAULIC FITTINGS

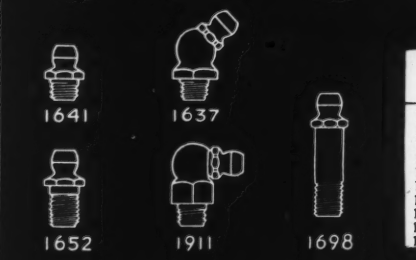
Part No.	A	B	C	D	E†	F	G	H
1606	1 13/16″	1″	3/8″	13/32″	1/8″ P.T.	7/16″	15/32″	90°
1611	1 1/64″	17/64″	13/64″	25/64″	1/8″ P.T.	7/16″	21/64″	30°
1612	53/64″	17/64″	13/64″	25/64″	1/8″ P.T.	7/16″	1/2″	67 1/2°
1613	49/64″	17/64″	13/64″	25/64″	1/8″ P.T.	7/16″	33/64″	85°
1614	1 5/64″	5/16″	1/4″	13/32″	1/8″ P.T.	7/16″	31/64″	105°
1620	1 5/16″	13/32″	5/16″	7/16″	1/8″ P.T. (Female)	7/16″	17/32″	45°
1623	1 3/16″	9/16″	1/4″	13/32″	1/8″ P.T.	7/16″	31/64″	67 1/2°
1638	2 9/32″	1″	3/8″	13/32″	1/8″ P.T.	7/16″	21/64″	30°
1649	2 25/32″	2 3/16″	5/16″	13/32″	1/8″ P.T.	3/8″	29/64″	67 1/2°
*1651	1″	3/8″	19/64″	13/32″	1/8″ N.P.T.F.	7/16″	1/2″	67 1/2°
1688	61/64″	17/64″	13/64″	25/64″	1/8″ P.T.	7/16″	27/64″	45°
*1692	1 7/32″	3/8″	19/64″	13/32″	1/8″ N.P.T.F.	7/16″	21/64″	30°
*1693	15/16″	3/8″	19/64″	13/32″	1/8″ N.P.T.F.	7/16″	31/64″	90°
*1950	1 1/8″	3/8″	19/64″	13/32″	1/8″ N.P.T.F.	7/16″	15/16″	105°

*Leak-proof seal. †Thread size listed as 1/8″ P.T. is 1/8″ P.T.F.—SAE EXTRA SHORT

1/4″ PIPE THREAD HYDRAULIC FITTINGS

Part No.	A	B	C	D	E†	F	G	H
1627	59/64″	23/64″	9/32″	17/32″	1/4″ P.T.	9/16″	.....	.....
1629	1″	11/32″	9/32″	17/32″	1/4″ P.T.	9/16″	5/8″	67 1/2°
1691	1 31/64″	Female Thread			1/4″ P.T.	5/8″	.....	.....

†Thread size listed as 1/4″ P.T. is 1/4″ P.T.F.—SAE EXTRA SHORT.



1/4″—28 THREAD HYDRAULIC FITTINGS

Part No.	A	B	C	D	E	F	G	H
1636	31/32″	19/64″	1/4″	17/64″	1/4″-28 Taper USF-Thd.	3/8″	3/8″	45°
1637	27/32″	11/64″	1/4″	1/4″	1/4″-28 Taper USF-Thd.	3/8″	3/8″	45°
1641	31/64″	11/64″	1/4″	1/4″	1/4″-28 Taper USF-Thd.	5/16″	.....	.....
1652	43/64″	23/64″	1/4″	1/4″	1/4″-28 Taper USF-Thd.	5/16″	.....	.....
1680	59/64″	39/64″	1/4″	17/64″	1/4″-28 Taper USF-Thd.	5/16″	.....	.....
1698	1 7/64″	51/64″	1/4″	17/64″	1/4″-28 Taper USF-Thd.	17/64″	.....	.....
1703	1/2″	7/32″	3/16″	1/4″	1/4″-28 Taper USF-Thd.	17/64″	.....	.....
1770-A	19/32″	1/8″	7/64″	1/4″	1/4″-28 NF-2-Thd.	3/8″	25/64″	45°
1792	31/64″	9/64″	1/8″	1/4″	1/4″-28 NF-2-Thd.	17/64″	.....	.....
1911	47/64″	11/64″	1/4″	1/4″	1/4″-28 Taper USF-Thd.	3/8″	15/32″	90°

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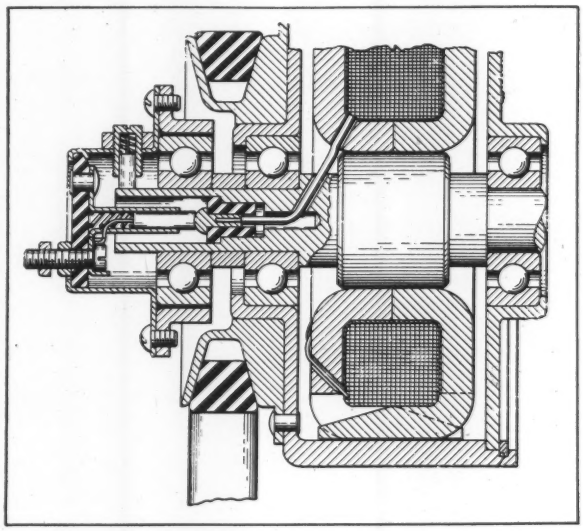
tained through action of a spring-loaded ball against the V-shaped lower end of the pawl. The drive is reversed by pivoting the pawl so that its other V surface bears on the ball, driving action then being through the other pawl tooth.

ELIMINATION OF SLIP RINGS in electromagnetic devices such as eddy current type clutches is accomplished by introducing the coil current through a contact button at the center of the rotor shaft. Patent

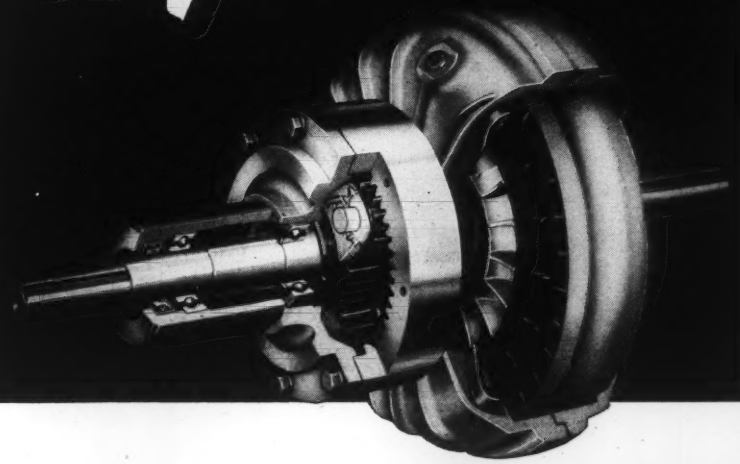


2,497,402, granted to Howard J. Findley and assigned to Eaton Mfg. Co., shows an arrangement whereby a spring-loaded conductor on the stationary housing contacts a conducting button at the center of the rotating shaft, resulting in a shorter, more direct path for current flow. In this application, ground connection for the magnetic coil is through a spring-loaded brush operating radially on one end of the rotor shaft. Energization of the coil locks the belt-driven outer drum to the output shaft.

CONSTANT VOLUME of pilot ignition oil is supplied to dual-fuel diesel engines, regardless of the speed or load of the engine, with the fuel injection pump covered in patent 2,502,989. Passages in the plunger type pump body are both closed for only a small part of the plunger pumping stroke, during which time oil is directed to the cylinder injector nozzles.

Depending on the engine requirements as interpreted by the governor, additional plunger travel either pumps more oil to the cylinder injector nozzles or bypasses oil back to the supply tanks. If the two lands on the plunger head are at the same angle from the barrel axis, rotating the plunger changes the injection timing without varying the volume of pilot ignition oil. If the plunger faces are made at an

Twin Disc Announces the Hydro-Wynd for Constant Tension Wind-up



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See your Twin Disc Dealer for full information about the new Hydro-Wynd for *your* winding application, whether it's wire, plastic, thread, paper or yarn. Currently available Hydro-Wynd units, singly or in compound, are capable of handling up to 7.5 horsepower.

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5 to 8 Pole 5-9/16″ w. x 3-5/8″ h. x 3-5/16″ d.

Reversing

2 to 4 Pole 5-9/16″ w. x 3-5/8″ h. x 3-5/16″ d.

Note: 10 and 15 ampere contactors have same mounting and overall dimensions.

ACCESSIBILITY

To replace contacts, it is not necessary to disassemble the complete contactor. Just remove the parts comprising the stationary and movable contacts. Contacts can be replaced without disturbing wiring. To change coil, remove magnet frame and coil assembly only. (See illustration below.)

FLEXIBILITY

Using a screw driver only, you can easily change any pole from normally open to normally closed. No special parts required. 10 and 15 ampere parts are interchangeable.

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Laboratory tests involving millions of operations, plus field service of thousands of R-B-M contactors on door operators, radio transmitters, packaging and weighing machinery, hoists, machine tools and many other industrial applications offer proof of dependable, trouble-free performance.

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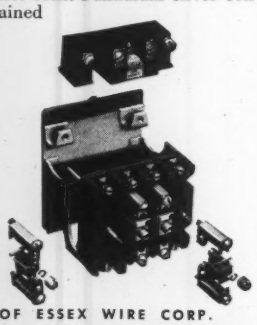

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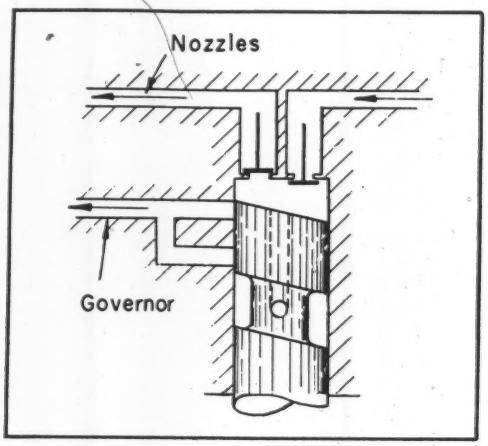
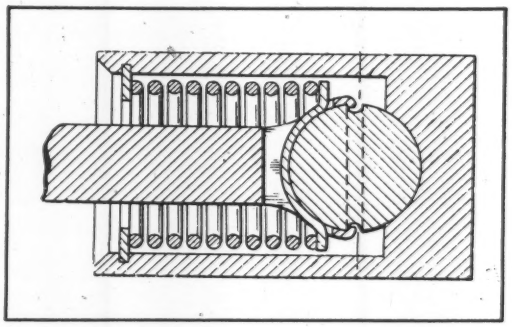
angle to the barrel axis but not parallel to one another, both timing and pilot volume are changed by rotating the plunger. However, if the upper face of the plunger head is perpendicular to the barrel axis,



rotating the plunger changes the volume pumped but timing remains constant. Edward Rathbun has assigned the patent to the Rathbun-Jones Engineering Co.

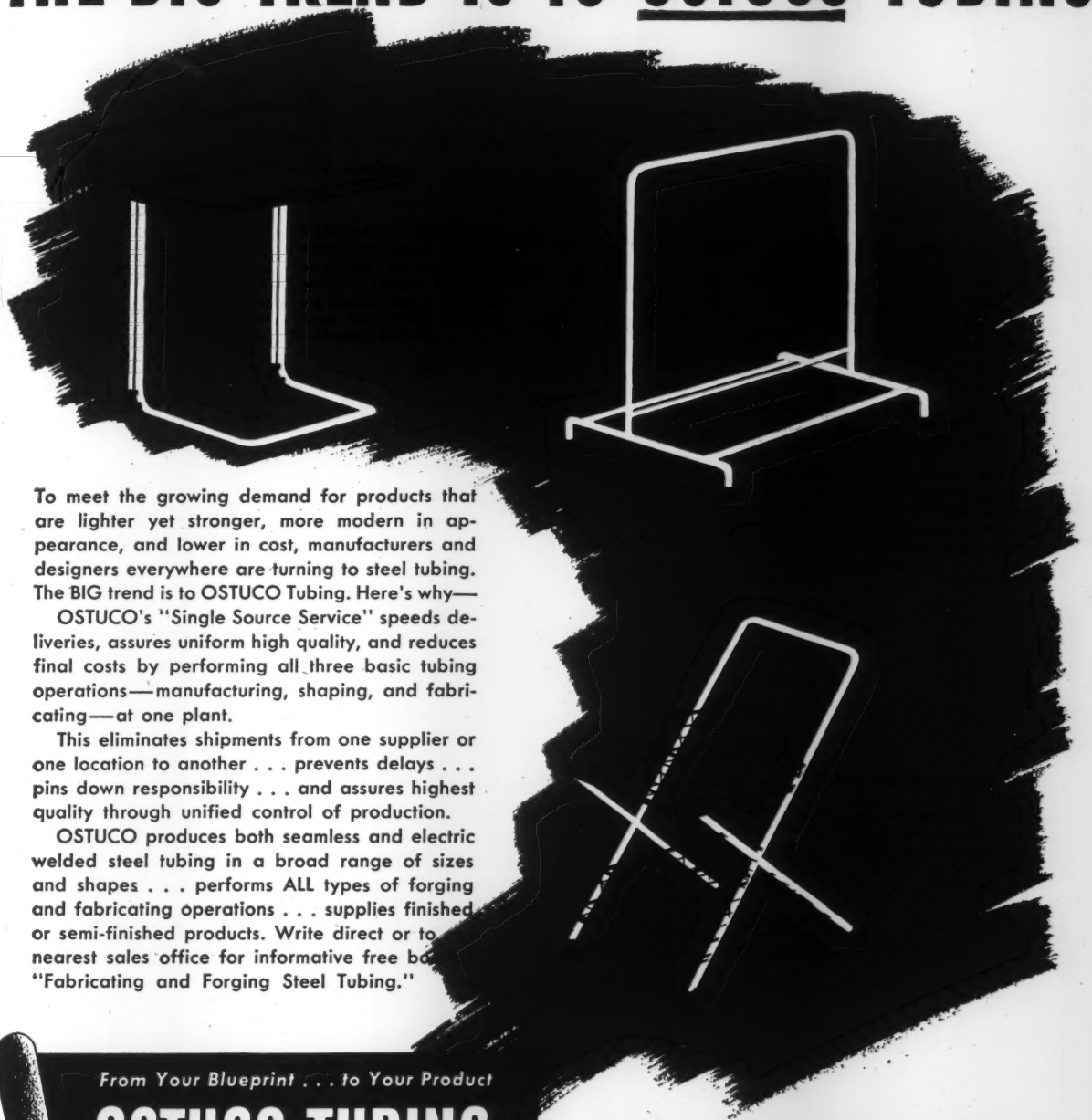
UNIVERSAL ACTION between the piston and connecting rod in refrigerator compressors is permitted by a ball-and-socket construction described in patent 2,500,669. Instead of the conventional wrist pin arrangement, the piston has a spherical socket which receives the ball-shaped end of the connecting rod. Piston and rod are held together by a coil spring



compressed and restrained inside the piston skirt by a split retaining ring. This arrangement eliminates play in the piston and rod assembly and decreases the degree of precision required in mating parts. Nash-Kelvinator Corp. has been assigned the patent by Ralph W. Doeg.

Complete printed copies of all patents are available from the Commissioner of Patents, Washington 25, D. C., for 25 cents each.

THE BIG TREND IS TO OSTUCO TUBING



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Design and Production

Edited by Colin Carmichael, editor of MACHINE DESIGN*;* 1611 *pages,* 5⅝ *by* 8⅝ *inches, cloth-bound; available through* MACHINE DESIGN, $8.50 *postpaid.*

This volume is directed primarily to the engineers concerned with the design and manufacture of machinery, appliances, mechanical equipment and other engineered products. Almost completely rewritten and about 300 pages longer than the previous edition, this volume presents coverage of: Selection of materials, design principles, design and selection of machine components, production processes, production plant equipment, and mathematical data.

Material new to this edition is: Data on thermoplastic, thermosetting and laminated plastics; plastics molding methods; precision investment casting; powder metallurgy; shot peening; brazing; application and selection of brazing alloys; machine hydraulics; seals; servomechanisms; industrial electronic control; and measurement and designation of surface quality.

Power

Edited by J. Kenneth Salisbury, general engineering and consulting laboratory, General Electric Co.; 1409 *pages,* 5⅝ *by* 8⅝ *inches, cloth-bound; available through* MACHINE DESIGN, $8.50 *postpaid.*

Intended for the engineer who is concerned with power, its availability, conversion and application, this volume covers: Basic components of power processes—air, fuel, heat, and steam; service functions, pumping and piping; power producing equipment—boilers, turbines, condensers, and gas turbines; refrigeration, heating, ventilating, and air conditioning; power as applied to transportation; electric power; atomic energy; instrumentation; power test codes;

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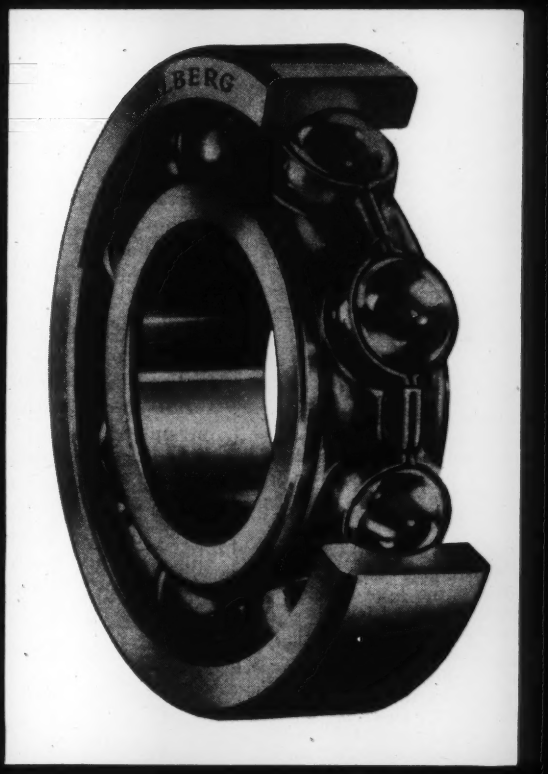

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□ □ □

Photoelasticity—Principles & Methods

By H. T. Jessop, senior lecturer, department of civil and municipal engineering, and F. C. Harris, lecturer, department of mathematics, University College, London; published by Dover Publications Inc., New York; 184 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN, $3.50 *postpaid.*

In order to design a product to best advantage, the design engineer needs to know how the stresses will be distributed in it under any given load system and also how the distribution will be affected by modification in shape. Within the last few years, photoelasticity has advanced to the point where it can serve in many cases to predict product behavior under stresses.

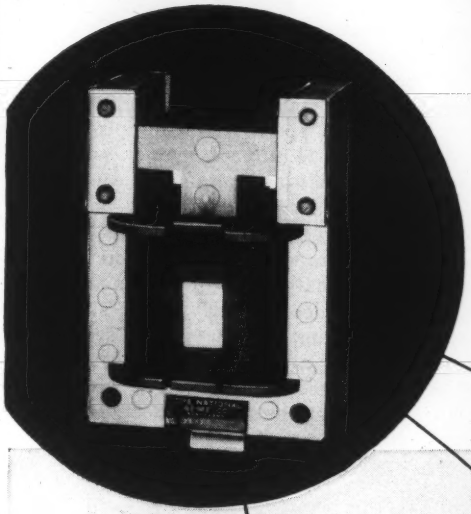
This book gives a detailed description of recent developments in both two- and three-dimensional stress analysis techniques. Covered in the book are: Stresses, strain and stress-strain relations; polarization and double refractions; theory of photoelasticity; reduction and interpretation of observations; stresses in three dimensions; and photoelastic apparatus, material and procedure. Featured is a special chapter on optics, giving both the theory necessary for a complete understanding of the subject and the practical details that arise from actual experimentation with photoelasticity. There is a separate chapter devoted to case histories from the authors' work. In the appendix are found examples of the mathematics involved in analyses where 'two dimension' conditions no longer apply.

□ □ □

High-Frequency Induction Heating
Second Edition

By Frank W. Curtis, consulting manufacturing engineer; published by the McGraw-Hill Book Co. Inc., New York; 389 *pages,* 6 *by* 9 *inches, clothbound; available through* MACHINE DESIGN, $6.00 *postpaid.*

This book is, in general, a practical explanation of the increasingly important technique of applying heat in the production of metal parts. The text covers



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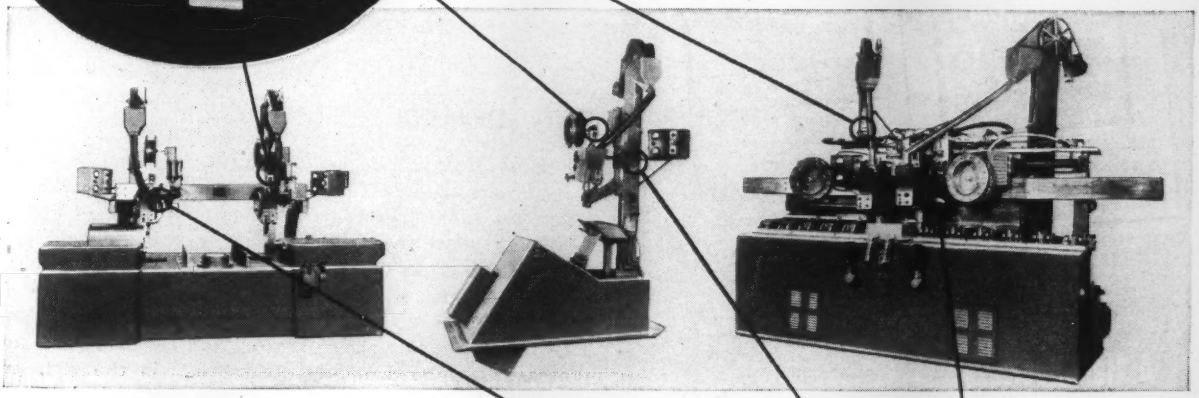

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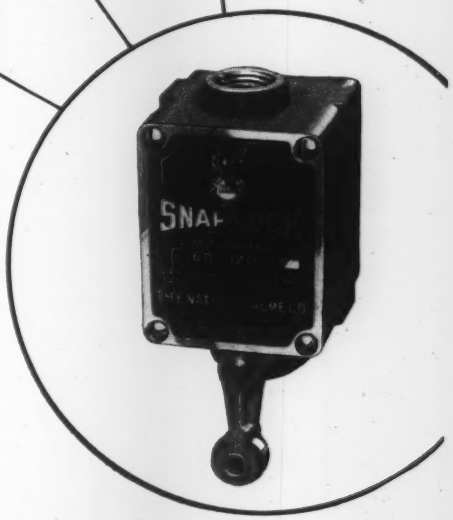
For complete information on standard Snap-Lock Limit Switches, ask for Bulletin EM-47.



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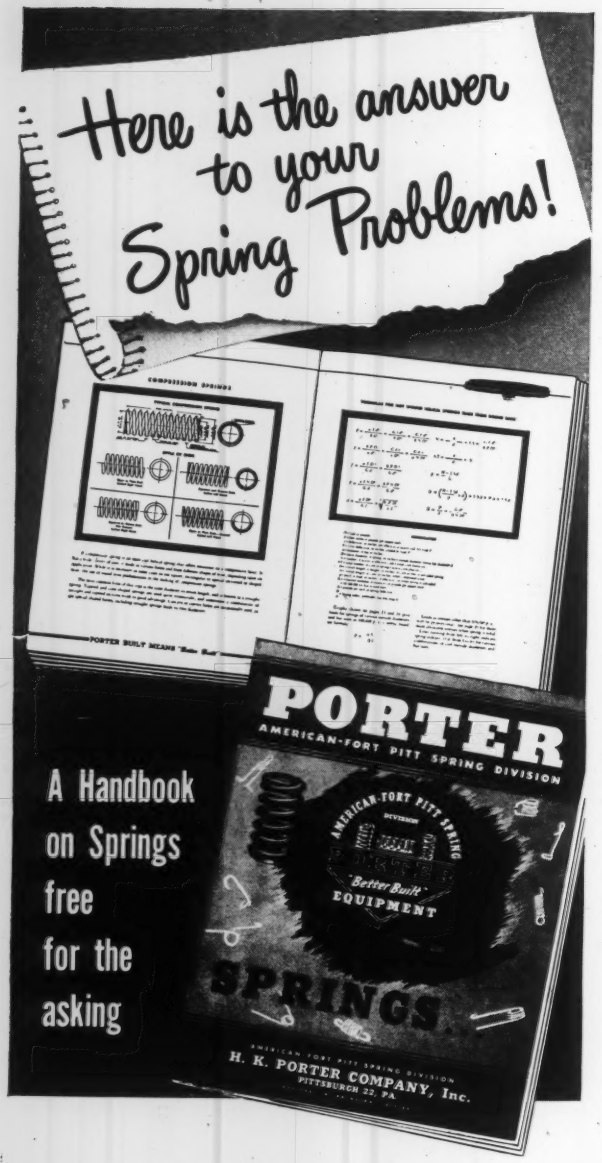
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the fundamental electrical principles involved in high-frequency induction heating; defines its range and scope; describes construction design of induction-heating coils; and discusses the many applications of induction heating to specific heating problems—hardening metal parts, joining of metal assemblies, drying painted parts, melting metals for centrifugal castings, applying heat for straightening, and other operations.

In the way of revision, this second edition includes a more comprehensive analysis of coil design and timely material on fixture designs. Types of steel best suited to induction heating, various means of quenching soft solders, brazing alloys, and fluxes are treated in full. Of special interest to designers is the chapter which deals with suggestions for product design best suited to induction heating.

New Standards

*High-Strength High-Temperature Internal Wrenching Bolts B*18.8—1950: This standard is intended for use in high-strength applications such as steam turbine work where the fasteners are subjected to high temperatures, 800 to 900 F, for long periods. These fasteners, compared to standard socket head screws in general use, have larger head proportions to provide greater bearing area and to insure greater strength in head and wrenches than in the body or threaded portions. The bolts have cylindrical heads to permit spot peening of the top of the head in counterbore where this method of locking is desired. Copies of this standard may be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N. Y. Price per copy of the standard is $0.35.

Manufacturer & Association Publications

Casting Design as Influenced by Foundry Practice: This 62-page, 8½ by 11-inch book is devoted to the design of Meehanite castings. Of particular value to the designer is the 29-page section of rules to be observed in designing castings to obtain maximum utility and production economy. For proper selection of casting materials twenty-four Meehanite metals are typed according to four casting classifications—general engineering, heat resisting, abrasion and wear resisting, and corrosion resisting. Properties data and heat treating procedure are presented. To give designers the rudiments of foundry practice, two sections of this book deal tersely with pattern making, molding and core making. Copies of the book may be obtained from the Meehanite Metal Corp., New Rochelle, N. Y., for $1.65 each.

Manual of Electric Instruments: Operating principles and application of instruments commonly used for electrical measurement are discussed and well illustrated in this 8½ by 11½-inch, 140-page book. The presentation is divided into thirteen chapters, each dealing with a particular instrument classifica-

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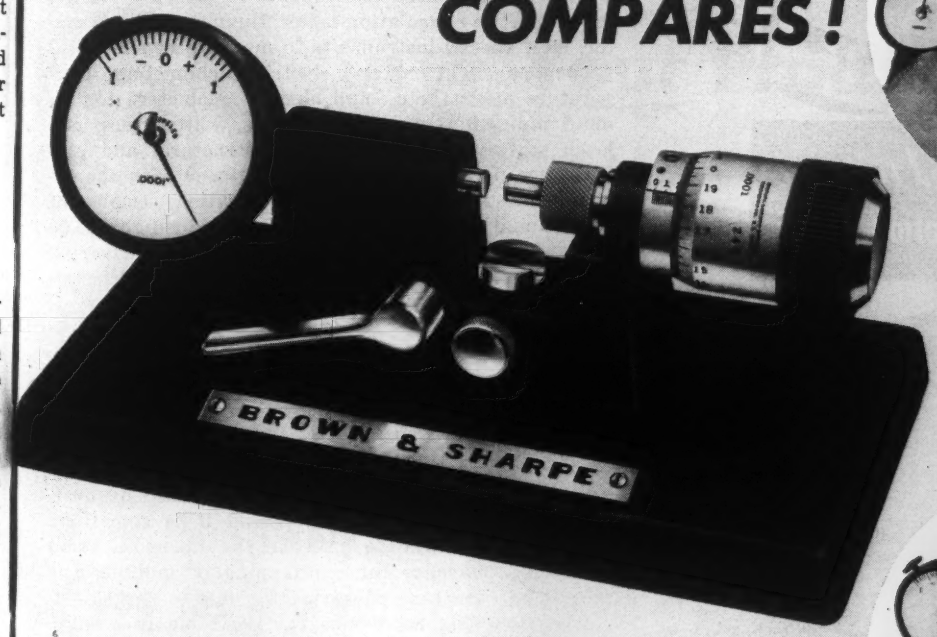

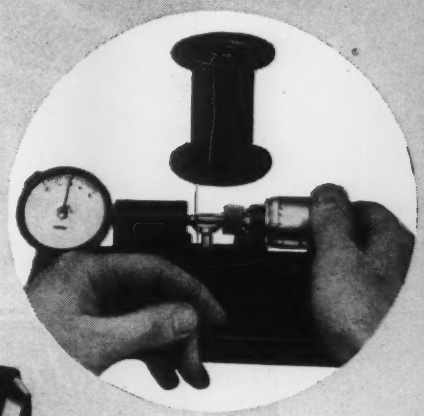

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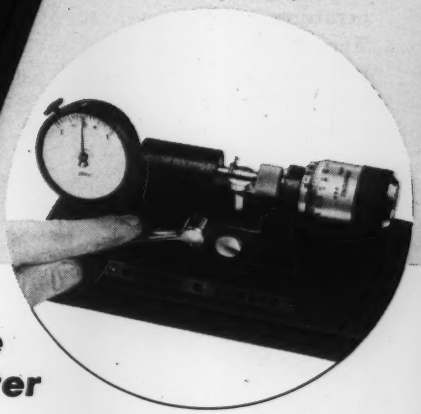
COMPARES
A finger operated lever is the only motion required for multiple inspection of duplicate parts.



No. 245 *The New Brown & Sharpe Indicating Bench Micrometer*

Here's a sensational new development that has no counterpart in the field of measuring devices! It's the new Brown & Sharpe Indicating Bench Micrometer No. 245 with original features — exclusive advantages — that serves multiple functions:

MEASURES to an accuracy of .0001″ over range of 0 to ½″. Reads directly from the thimble. Or, for fast setting and measuring, can be set to nearest thousandth and then read in increments of .0001″ on dial gage.

COMPARES . . . for multiple inspection of small parts. Dial indicator gives direct readings of variations from standard sample, in ten-thousandths. No need for additional standards.

What's more, it's fast! A slight, short movement of the anvil retractor lever permits instant insertion or removal of work pieces.

Measuring surfaces of the No. 245 are tipped with tungsten-carbide for minimum wear. Anvil movement is in straight line — frictionless — with measuring surfaces parallel in all positions. Adjustable work support facilitates centering. Measuring pressure is constant and adjustable — factory-set at any pressure you specify, from 8 oz. to 2 lb.

See the new No. 245 Bench Micrometer at your distributor's, or write today for illustrated bulletin. Brown & Sharpe Mfg. Co., Providence 1, R. I., U.S.A.

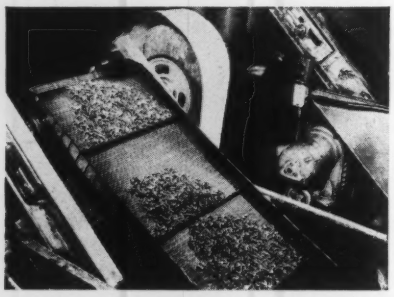
We urge buying through the Distributor

BROWN & SHARPE B·S

Your Cambridge woven wire conveyor belt is engineered from the raw materials to the finished belt to give you maximum efficiency with a minimum of attention and maintenance.

Cambridge designs withstand heavy loads, high temperatures, cold, corrosive compounds or wet processes. They may be fabricated of any metal or alloy in any mesh and weave. Where necessary, Cambridge belts can be fitted with various types of selvages, cleats and drives.

Call your Cambridge Field Engineer whenever you have a problem concerning movement of materials during processing.



A Cambridge Rod-Reinforced Belt with Plate Selvages and Fabric Cleats provides cost-cutting continuous quenching of screws, nuts and bolts. This belt design offers the highest tensile strength, low thermal capacity and successful operations in temperatures as high as 2100° F.

VALUABLE BOOK, FREE! A complete, basic reference on conveyor belts, designs, systems and metallurgy. Write for your copy or call your nearest Cambridge office.

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Wire cloth in rolls. | Also specialized wire fabrications

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tion: Permanent-magnet moving-coil instruments for d-c measurements; rectifier instruments for high-frequency measurement; thermocouple instruments for high-frequency measurement; moving-iron ammeters and voltmeters for a-c measurement; electrodynamometer instruments—wattmeters and varmeters; power factor meters; synchroscopes; frequency meters; recording instruments; specialized instruments to measure electrical quantities—d-c galvanometers, magnetic oscillographs, electrostatic voltmeters, etc.; specialized instruments to measure electrical properties—insulation tester, fluxmeter, Gauss meter, etc.; special instruments to measure nonelectrical properties—light meters, electric tachometers, temperature meters, etc.; and electric telemeters for remote indication of amperes, volts, watts, vars, etc. Each section contains sufficient structural and performance data to acquaint the engineer with the instruments, their design and application. Copies of the manual (GET-1087A) can be obtained for $1.00 each by writing to the Distribution Center, Advertising & Sales Promotion Divisions, General Electric Co., Schenectady 5, N. Y.

High-Speed Cams

(*Concluded from Page* 111)

deserves particular attention. Every effort should be taken to make the cam motion period as long as possible. The period can often be increased by overlapping successive movements, and it is sometimes actually worthwhile to increase the throw in order to secure clearance for overlapping of motions and permit the increase of period.

The following conditions are desirable in a high-speed cam drive to insure smooth operation and minimum maintenance:

1. A maximum pressure angle less than 30 degrees
2. A properly selected cam profile
3. An accurately machined cam profile
4. A hardened cam roller of adequate diameter and width plus a cam material of adequate hardness to resist wear, mushrooming, and fatigue
5. Adequate lubrication
6. In a spring-loaded cam system, adequate spring-loading to maintain follower contact at all times plus proper spring design to prevent surging
7. Minimum backlash at all points with bearings properly selected to minimize wear
8. Minimum possible driven mass
9. Minimum possible throw
10. Maximum possible period of operation
11. Maximum practical rigidity in cam supports and cam drive and in the coupling between cam follower and driven mass.

REFERENCES

1. Franklin de Konde Furman—*Cams, Elementary and Advanced*, John Wiley, 1921.
2. G. T. Chapman—"Parabolic Cams Smooth Velocity Curves," MACHINE DESIGN, Feb. 1936, Pages 39-40.
3. Wade H. Shorter Jr.—"Cam Design Analysis," *Product Engineering*, May 1940, Pages 223-227 and June 1940, Pages 264-265.
4. Winston M. Dudley—"A New Approach to Cam Design," MACHINE DESIGN, July 1947, Pages 143-148 and 184.
5. John A. Hrones—"Key Factors in Cam Design and Application," MACHINE DESIGN, April 1949, Pages 127-132; May 1949, Pages 107-111 and 178; June 1949, Pages 124-126.
6. D. B. Mitchell—"Cam Follower Dynamics," MACHINE DESIGN, June 1950.

Buried Treasure

in Your Product?

More than likely... if you can dig up New Ways to make it do more for your customers ...by COUNTING

No. 1239 Predetermining Counter signals operator or actuates mechanism to stop machine at end of pre-set run.

Dig deeply into this million-dollar question: "How could *my* product increase its usefulness and sales . . . *by counting?*" And you may well uncover a new and distinctive merchandising appeal that will set your product apart from competition . . . as so many manufacturers have done.

It's as simple as this: If your product is mechanically or electrically operated, then it's definitely worth a search to see if there's hidden sales-treasure buried there. This can be quickly determined by some fast spade-work done by a Veeder-Root engineer, paired off with your design engineer. And the digging can get under way . . . *any time you say.*

Write for 8-page "Counter Book" which shows all types of V-R electrical, mechanical, and manual counters . . . standard and special.

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Veeder-Root COUNTERS

Here's why G-E Silicone Rubber was specified

for this bushings application

Naval radar makes exacting demands upon its component parts. That's why self-sealing General Electric silicone rubber was specified for the bushings of this special-purpose transformer built by the Raytheon Manufacturing Company, Waltham, Mass. Inserted through punched round holes, the bushings seal hermetically by compression. They retain elasticity from −55 F to 520 F, absorb shocks that would shatter glass, reduce cost per terminal, and show no tracking characteristics. **Resilient G-E silicone rubber** won't stick to hot metal surfaces when clamped under compression. It contains no plasticizers that might affect electrical properties, and contaminating adhesives are not required.

Look for G-E silicone rubber in the parts you buy. Specify it when you order. It's ideal for bushings, gaskets, heat sealers and shock absorbers. For more information on G-E silicone rubber, write to Section Z2, Chemical Department, General Electric Company, Pittsfield, Mass. (In Canada: Canadian General Electric, Toronto.)

SILICONE RUBBER

GENERAL ELECTRIC

Locating Axis of Rotation

(*Concluded from Page* 140)

if z is set equal to zero the projection on the xy plane is obtained.

It can be seen from the foregoing that by judicious choice of the co-ordinate planes, the equation of the axis can be found for one of the views in the layout. The equation will be in the form $y = mx + b$.

The problem presented in *Fig.* 2 can be solved easily if the co-ordinate system is designed using frame C as the xz plane shown in *Fig.* 4.

It is also an easy matter to check the calculations by this method. If a value of x, y or z is assumed, the co-ordinates of a third point C, on the axis of rotation, can be found. The true distance between C and A should equal the true distance between C and A'. If the point C has co-ordinates $x_c y_c z_c$ the distance AC will equal:

$$AC = (x_c - x_1)^2 + (y_c - y_1)^2 + (z_c - z_1)^2$$

The distance $A'C$ will equal:

$$A'C = (x_c - x_2)^2 + (y_c - y_2)^2 + (z_c - z_2)^2$$

In solving a particular problem, the first step is to establish a co-ordinate system. Careful choice of this system will many times simplify the solution. In almost any problem it will be necessary to do some juggling before the final satisfactory design is obtained. If the co-ordinate system is set up so the probable juggling will occur along one of the axes of the system the amount of work required to reach a final solution is reduced considerably.

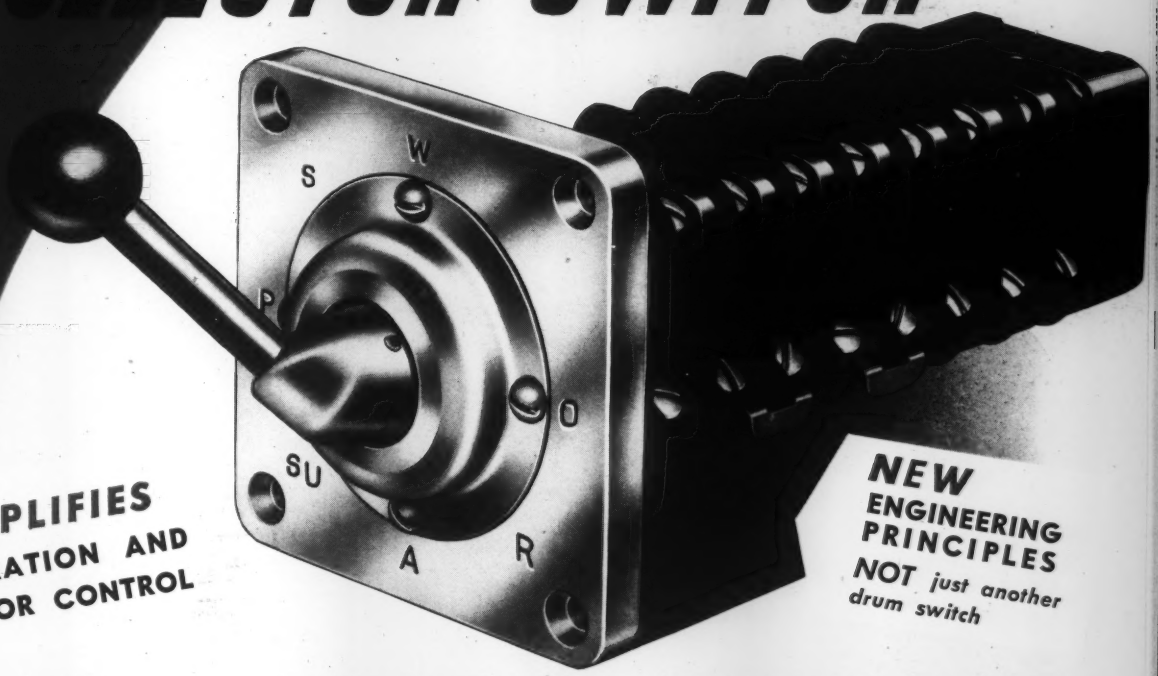
Considering the aircraft landing gear example, for purpose of calculation two points are chosen on the center line of the axle of the wheel; the co-ordinates of these two points, both in the extended and retracted position, are then calculated. Once the co-ordinates have been calculated it is only necessary to subsitute them in the proper equations and then solve for the equations of the projections of the axis in the co-ordinate planes. The equations of the projections will be in the form $y = mx + b$. From the three equations obtained in this way it is possible to determine the point at which the axis pierces the supporting structure, *Fig.* 2*b*. Once these points have been determined an investigation as to the possibility of designing adequate structure at these points can be made. If the axis is not in a convenient position, one or both of the limiting positions (extended or retracted position of the wheel) must be varied slightly, or the axis moved and the resultant change in position of the wheel calculated.

Using the foregoing information and the equations of the shop jigging or banking planes it is possible to calculate true angles and offsets for the machining jigs and fixtures.

Although this discussion has been concerned primarily with an aircraft problem, it can be seen that the same approach can be applied to any problem where it is necessary to locate an axis about which a rigid member or assembly must rotate.

The New PUSH-PULL SELECTOR SWITCH

ARROW AH HART



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NEW ENGINEERING PRINCIPLES
NOT just another drum switch

CUSTOM BUILT TO SUIT YOUR NEEDS, THIS REVOLUTIONARY NEW SWITCH CUTS YOUR CONTROL COSTS, TOO!

The new Arrow-Hart PUSH-PULL-SELECTOR Switch provides a single point of control for a multiple-operation machine. Any combination of momentary contact elements can be arranged — to act in place of push buttons, relays and contactors to start or stop actions. Up to 16 dial positions are available. You can stop or start in any position. You can move from any position on the dial to any other position — without activating any of the intervening operations. Control and load circuits may be combined in as many sections as you require. Each section has 2 completely separate circuits; each can be opened or closed independently in all positions. Any combination required can be used in any position—3-wire or 2-wire control.

SEE PROOF OF THESE STATEMENTS ON NEXT PAGE

CONTROLS and APPARATUS FOR *Outstanding* PERFORMANCE

ARROW AH HART

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THE ARROW-HART & HEGEMAN ELECTRIC COMPANY
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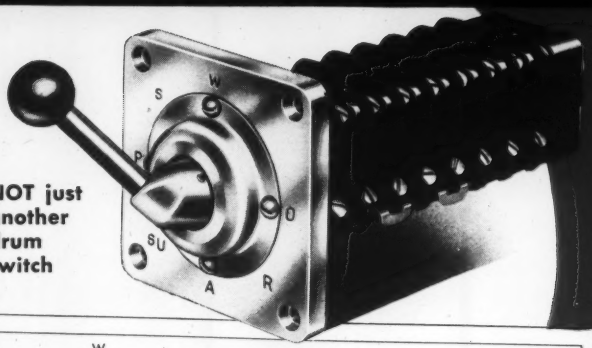
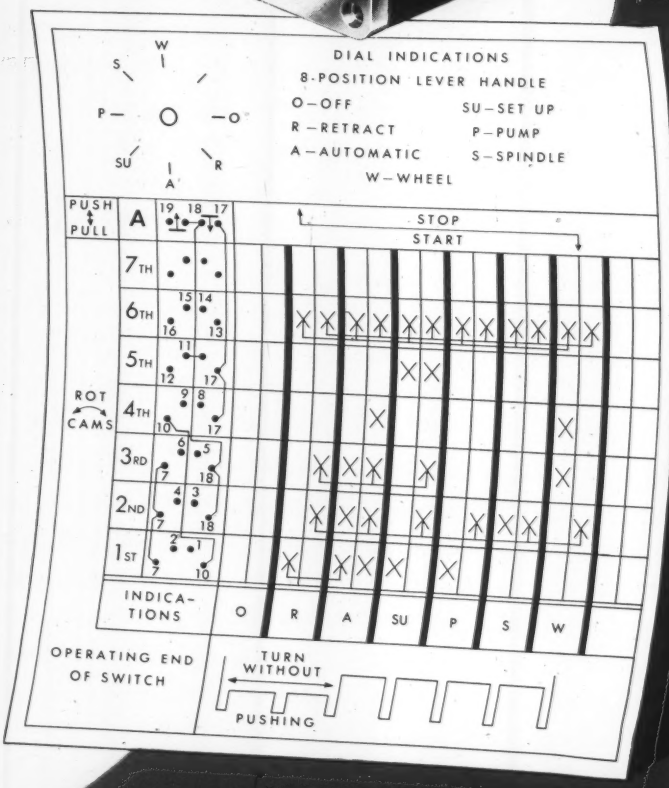
SAVE TIME, MONEY AND SPACE with

The New PUSH-PULL SELECTOR SWITCH

ARROW AH HART



NOT just another drum switch



Arrow-Hart PUSH-PULL-SELECTOR Switches can effect unheard-of savings in your electrical control costs. You save installation and operation time through simple, central control of multiple operations. You save money through lower initial costs and elimination of many components. And you benefit from versatility through the wide selection of circuit-positions. Safe, foolproof operation from one point of control.

The target sheet illustrated at right for the operator's control "PPS" Switch displays the legend and circuit requirements for the New Departure Ball Race Grinder. 2 "PPS" Switches and 3 starters provide complete control. Without the "PPS" Switches control could not have been effected with less than 10 push buttons and 7 starters. Savings in initial costs, installation and operation time speak for themselves!

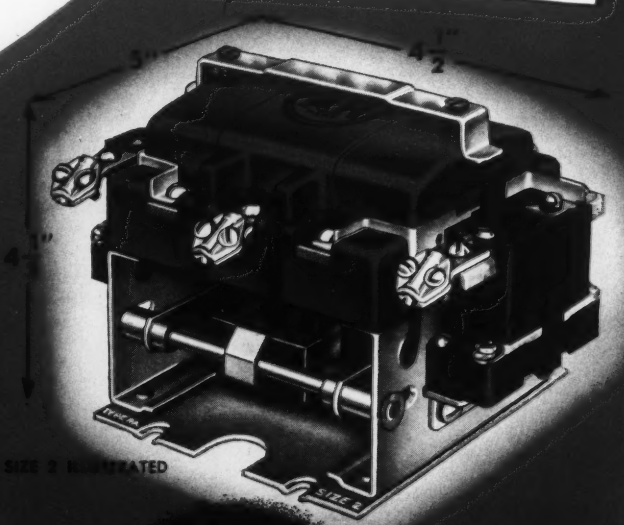
CUSTOM BUILT TO YOUR SPECIFICATIONS

Arrow-Hart PUSH-PULL-SELECTOR Switches are custom built — designed to your individual specifications by Arrow-Hart engineers. The New Departure application of the "PPS" Switch increased production to better than 20%. Similar savings are available to you. Our design, production and assembly facilities are placed at your disposal — to give you the finest in electrical control equipment.

TELL US YOUR CONTROL PROBLEM — give us your requirements — and we'll design a control unit that will give you years of economical efficient and maintenance-free service. WRITE TODAY FOR "PPS" FOLDER and SPECIFICATIONS SHEETS.

ASK ABOUT THE NEW TYPE "RA" MAGNETIC STARTERS 1/3 size of Conventional Controls

They feature exclusive right angle balanced mechanism, larger contacts, Plaskon Alkyd hood and base. Easy to wire and service. Available in sizes 0, 1, 2 and 3 in local and remote control or selector switch types. Write for literature.



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Brake Designs

(*Concluded from Page* 150)

matical term combining all of the physical dimensions of the brake, exclusive of friction coefficient, which affect self-energizing. The actuation factor is the product of the actuation constant and the friction coefficient; it expresses that portion of total brake input forces supplied by the self-energizing feature.

For a given brake output, let P' = Brake application force without self-energizing and P = Brake application force with self-energizing. Then, by definition, the actuation factor

$$A = \frac{P' - P}{P'}$$

For the non-self-energizing condition, P' becomes the same as $P + F$ in the force polygon or

$$P' = \frac{F \tan \phi}{2\mu}$$

$$P = \frac{F}{2} \quad \frac{(\tan \phi - 1)}{\mu}$$

Solving the original equation for A gives

$$A = \frac{\mu}{\tan \phi}$$

and the actuation constant, Q, is

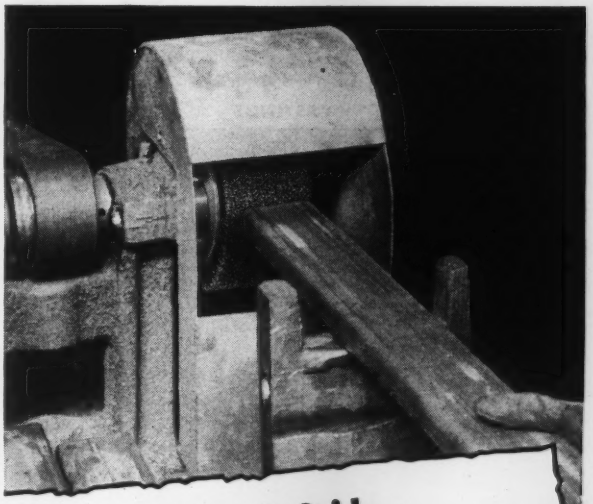
$$Q = \frac{1}{\tan \phi}$$

Hence for a ramp angle of 35 degrees, the actuation constant Q = 1.43, and with a coefficient of friction of 0.35, the actuation factor A is 0.50. For 32½ degrees, Q = 1.58 and A = 0.55. In the latter case, the brake is said to be 55 per cent self-energizing.

Because of the relatively low thermal and mechanical distortion in the disk brake housing, it is feasible to use a very simple self-adjuster to compensate for lining wear. Unlike a drum brake, the disk brake need not be compensated for thermal or mechanical deflections.

In the early experimental designs of the disk brake, friction lining was applied to the pressure plates in the form of a continuous ring, similar to clutch facings. However, difficulty was experienced with a tendency to score rapidly and wear the surface of the lining when there was no escape path for lining dust and foreign material. Under some conditions pieces of lining dust would "ball up" and completely tear the lining off in a few stops. After a series of tests to overcome this deficiency, it was found that applying the lining in 4-inch long segments not only overcame this difficulty, but also increased brake effectiveness and resistance to fade.

In addition to low pedal effort, high stability, and greatly increased heat absorption and dissipation capacity of the disk brake, it is possible that the combination of automatic wear adjuster and low wear rate of the lining may eliminate the necessity for periodic maintenance throughout normal car life.



National Electric Said...
"*Make it* STIFF! TOUGH! RUGGED!"

When National Electric Products Corp. put their brushing problems in the lap of Pittsburgh's skilled brush engineers, they requested a special brush for rough duty. It had to be **stiff** enough to remove burrs and scales... **tough** enough to penetrate and clean... **rugged** enough to stand up under heavy operation at high speed.

Pittsburgh engineers came up with the right answer in a special "power driven" brush that met each National Electric specification . . . also increased plant efficiency and stepped up production.

You Can Depend on PITTSBURGH to Solve Your Brush Problems

We have brushes of all types in the complete Pittsburgh line which should comply with your specific needs . . . whether you're in glass, steel or plastics . . . in paper, tires, shoes or automobiles . . . because there's a Pittsburgh brush for every industrial use.

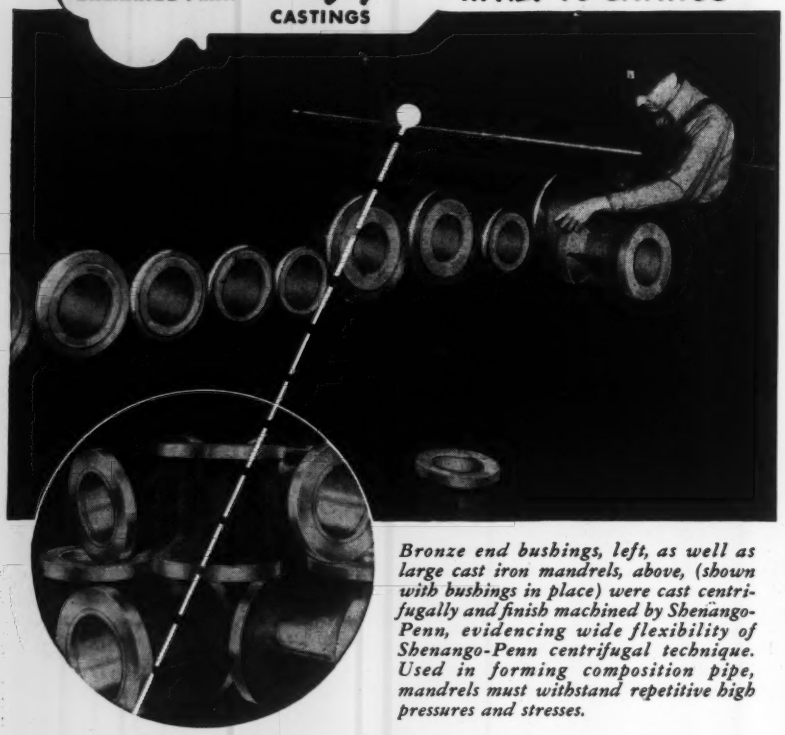
But if you have special problems, our staff of skilled brush engineers will be glad to design and build any type of power-driven brush to meet your particular specifications. What's more, they'll do it quickly, efficiently and economically.

Write or telephone PITTSBURGH PLATE GLASS COMPANY, *Brush Division*, Dept. W-1, 3221 Frederick Avenue, Baltimore 29, Md., for information . . . or tell us what your problem is, and let one of our engineers solve it for you. No obligation.

PITTSBURGH
Power Driven
BRUSHES

BRUSHES • PAINT • GLASS • CHEMICALS • PLASTICS
PITTSBURGH PLATE GLASS COMPANY

SHENANGO-PENN *Centrifugal* CASTINGS ... KEY TO SAVINGS



Bronze end bushings, left, as well as large cast iron mandrels, above, (shown with bushings in place) were cast centrifugally and finish machined by Shenango-Penn, evidencing wide flexibility of Shenango-Penn centrifugal technique. Used in forming composition pipe, mandrels must withstand repetitive high pressures and stresses.

For high pressures, high stresses . . .

IF you need rolls, or *any* symmetrical or annular parts requiring unusual strength, uniformity, wear resistance or freedom from porosity and defects, check with Shenango-Penn. These very qualities are inherent in the carefully controlled Shenango-Penn *centrifugal* casting process. You stand to save time and money . . . to avoid needless trouble.

Check also on Shenango-Penn's modern machining and finishing service. It combines complete facilities and years of specialized experience to meet your specifications with over-all precision . . . another time-and-money saver!

For free descriptive bulletins, for specific data on ferrous or nonferrous work, large or small, semi or finish machined, write to . . .

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MONEL METAL • NI-RESIST • MEEHANITE® METAL

SALES AND SERVICE PERSONNEL

THREE NEW district office managers have been appointed for Allis-Chalmers' general machinery division. They are **J. H. Burrus,** new manager of the Milwaukee district; **U. E. Sandelin,** Portland district; and **Stephen C. Bacon,** Seattle district. Mr. Burrus, manager of the Portland district since 1949, succeeds **H. A. Balding,** who is retiring from active management of the Milwaukee district after being associated with Allis-Chalmers since 1902. For several months, however, he will act as consultant and adviser to Mr. Burrus. Mr. Sandelin has been employed by the company since 1929 and was manager of the Seattle district for five years. Having joined the company in 1938, Mr. Bacon has been associated with the Seattle office for twelve years.

♦

Cecil E. Lucas of Lucas Screw Products Inc., Rochester, was elected president of the National Screw Machine Products Association at its recent annual meeting. **C. R. Wedler Jr.** of Wedler Bros. Inc., Cleveland, was elected a director of the association.

♦

Appointment of **Walter S. Howard** as director of engineering and development of automotive and high strength malleable castings has been announced by the Lake City Malleable Co., Cleveland. Until recently, Mr. Howard has been chief inspector and director of quality control for White Motor Co. of Cleveland.

♦

According to a recent announcement, the Trumbull Electric Mfg. Co. of Plainville, Conn., has put into effect a new plan of organization under which the company's marketing department will operate on a regional basis. Under this new plan, overall company marketing activities will be carried on in each of four geographical regions under the direction of regional managers. **Richard C. Lipps,** who has been with company for twenty years, has been made regional manager in the East, which includes field sales districts and sales engineering functions from Maine to Georgia. **Allan A. Watson** was appointed central regional manager, giving him responsibility for market-

Why M. W. Kellogg's drawings are reproduced on Kodagraph Autopositive Paper

In its London and Paris branches the M. W. Kellogg Company, largest designer and builder of oil refineries, keeps on hand intermediates of its home-office drawings and data sheets.

These are needed for daily reference . . . and to produce blueprints and direct-process prints for its European and Middle East operations.

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Dependable intermediates are a must! And M. W. Kellogg's home office (New York City) gets them promptly, at low cost, simply by ordering intermediates on Kodagraph Autopositive Paper.

Kodagraph Autopositive intermediates are photo-lasting in the files . . . will not fade, turn yellow, become brittle, or otherwise deteriorate. *Extra protection* which M. W. Kellogg appreciates . . . which forestalls cable "calls" for replacements . . . and costly tie-ups all along the line.

Kodagraph Autopositive intermediates are excellent print-making masters . . . have dense photographic black lines . . . on an evenly translucent, high quality, durable paper base. They do not smudge or lose line density in print production . . . and they turn out sharp, legible final prints time after time—at uniform, practical machine speeds.

Another advantage that saves time and dollars in the field. And another reason why M. W. Kellogg orders Kodagraph Autopositive intermediates of its "foreign work" drawings.

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"THE BIG NEW PLUS" in engineering drawing reproduction

- **It enables you, or your local blueprinter, to produce positive *photographic* intermediates directly . . . at a new low cost—using existing equipment and standard photographic solutions.**
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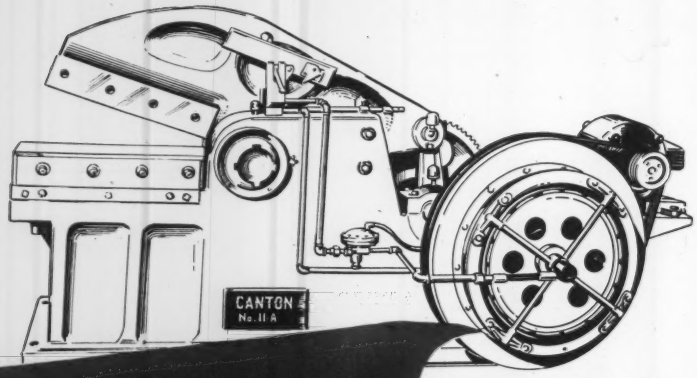
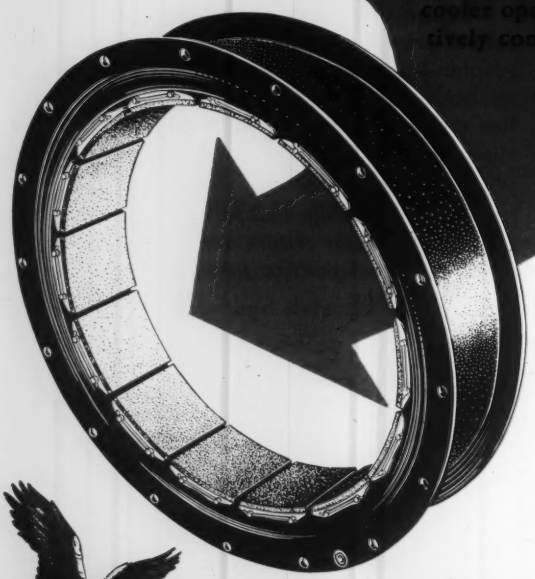


Hill Acme 11A Canton Shear Equipped with Fawick 18CB500 Clutch and 14E475 Brake.

Your Practical Answer

TO CLEAN SHEAR JOBS

• Alligator and other types of shears produce FULL power and extra-smooth shearing when equipped with Fawick Airflex Clutches. In addition, Fawick-equipped machines have a quick, shock-free response to finger-tip control. Only Fawick Clutches provide users with (1) 360° constant-velocity contact with the friction surface, (2) uniform-pressure axial contact between the full width of the friction surface and the drum, (3) shock absorbing flexibility of the torque-transmitting rubber tube, (4) open-type clutch construction assuring cooler operation, (5) sensitively controlled torque.



For complete technical information on Fawick Clutch and Brake Units, write to the Main Office, Cleveland, Ohio, for Bulletin No. 300.

All desirable clutch characteristics are built into Fawick Airflex units

FAWICK Airflex CO., INC.
9919 CLINTON ROAD CLEVELAND 11, OHIO

FAWICK CLUTCHES = + + + = PEAK EFFICIENCY
AXIAL CONTACT · 360° RADIAL CONTACT · SHOCK ABSORPTION · CONTROLLED TORQUE

ing activities in the Middle West and South, including field sales districts and sales engineering functions extending from western Pennsylvania to Colorado as far north as Canada. The company's marketing activities in the California region, including all of California and Arizona and part of Nevada, will be under the direction of **J. Warren Barry,** who has been promoted from his position as California operations manager. The Northwest region, covering all of Washington, Oregon and Idaho and parts of Nevada, Colorado and Montana, will be under the managership of **Willis I. Downie.**

♦

Sales manager of the plastics division of the B. F. Goodrich Co., **Clyde O. DeLong** has been named general manager of the division in charge of manufacturing and sales. Mr. DeLong, whose headquarters are in the company's plant in Marietta, O., joined B. F. Goodrich in 1928.

♦

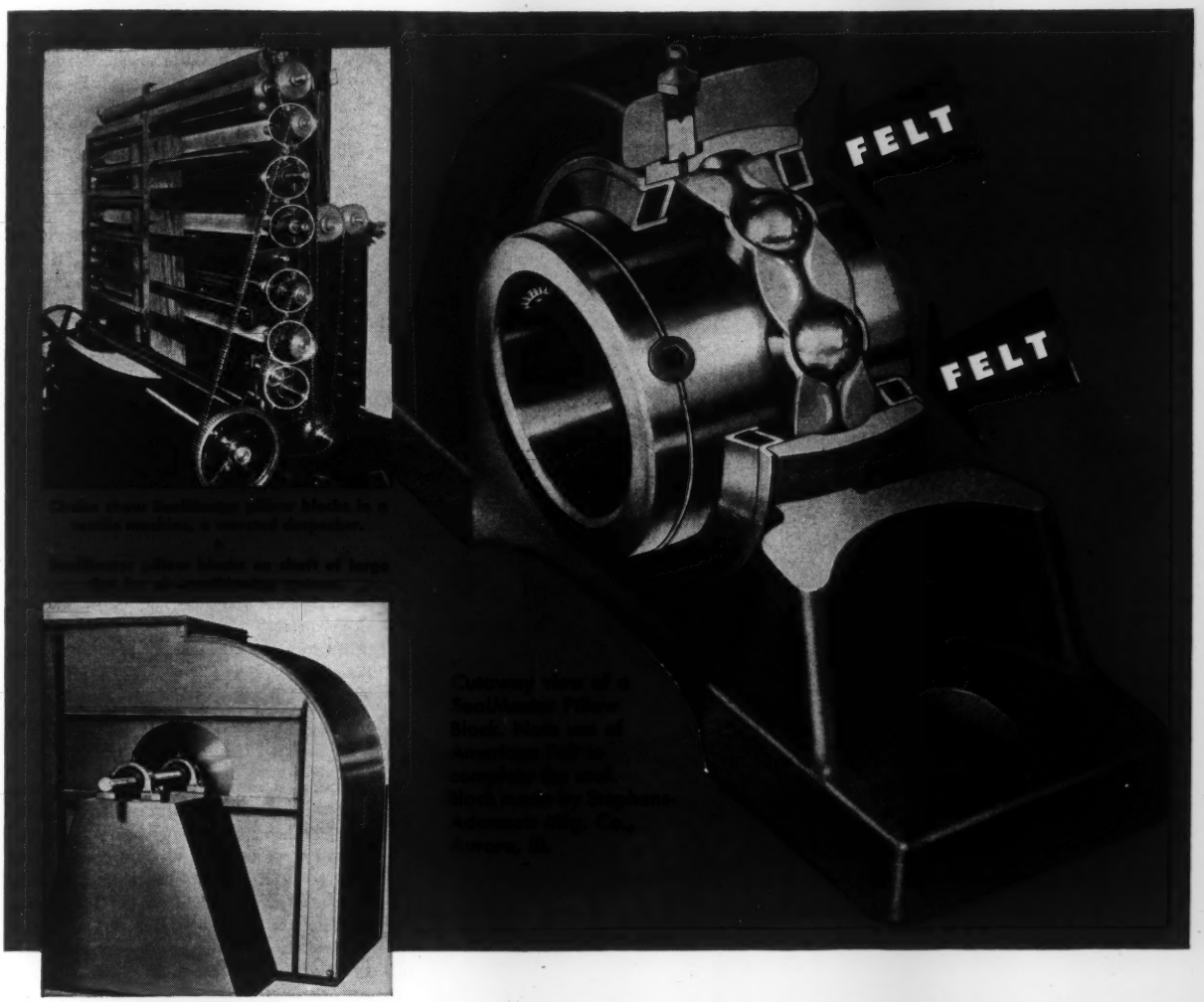
A. F. Boucher has assumed sales and engineering responsibilities as district manager in the Detroit area for the Lincoln Electric Co. of Cleveland. He was transferred to Detroit from Milwaukee where he has been district manager since 1947. Also, **I. R. Bartter** has been transferred from Grand Rapids, Mich., where he has been district manager since 1945, to Milwaukee to assume the responsibilities of district manager in that area.

♦

With headquarters in Minneapolis, **H. R. Comstock** has been appointed manager of the central division of Goodyear Tire and Rubber Co.'s mechanical goods sales, replacing **H. E. Langdon,** whose new duties have not been assigned. Central sales division is comprised of the Chicago, Detroit, Minneapolis and Cincinnati districts. Mr. Comstock has served as a salesman for many years, having joined the Goodyear company in 1929 as a general line salesman in Minneapolis. He entered mechanical goods sales six years later and was appointed mechanical goods district manager at Minneapolis in 1940.

♦

Having served as manager of customer service for the past year, **Joseph H. Harris** has been appointed sales manager of the North Chicago, Ill. branch of the Atlas Powder Co.'s industrial finishes department. **Avery Doolittle** succeeds Mr. Harris as manager of customer service. Zapon representation in Ohio will be augmented with the assignment of **Roy T. Riggs** to a territory in the northern part



Let's LOOK INTO the Use of FELT by SEALMASTER

• Here is an exceptionally interesting example of the use of American Felt in a bearing seal. Note that the seal consists of an inner labyrinth ring pressed into the outer race of the bearing, an outer flinger ring pressed onto the inner race of the bearing, and a ring of felt in the channel between the two. The felt rotates with the outer ring, and as it is assembled without pressure, there is no danger of glazing or wear.

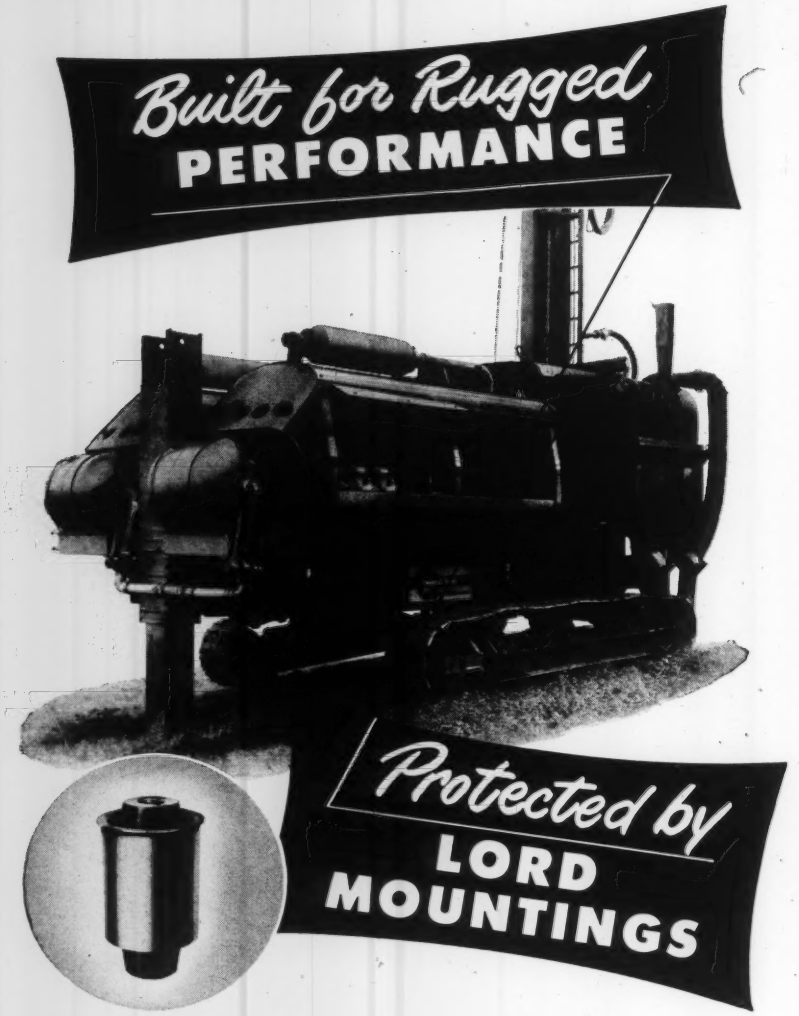
Now let's see what happens in service. Due to the rotation of the outer steel ring and the felt ring, a centrifugal action is developed. Three effects result from this action in combination with the design of the labyrinth: 1, entry of dirt into the seal is prevented; 2, excess grease that may work its way past the vortex or trap created by the steel seal ring on the outer race passes slowly through the felt; 3, the felt is kept clean and free from glazing. When rotation stops there are still the same barriers, except the centrifugal. No wonder these pillow blocks run for years with such protection!

• American provides various types of felt, plain or laminated with impervious materials, to meet the need for reliable seals. Write for authoritative Data Sheet No. 11, "Felt Seals, Their Design and Application", complete with illustrative samples.

American Felt Company

TRADE MARK

GENERAL OFFICES: 22 GLENVILLE RD., GLENVILLE, CONN. ENGINEERING AND RESEARCH LABORATORIES: GLENVILLE, CONN. —PLANTS: Glenville, Conn.; Franklin, Mass.; Newburgh, N. Y.; Detroit, Mich.; Westerly, R. I. — SALES OFFICES: New York, Boston, Chicago, Detroit, Cleveland, Rochester, Philadelphia, St. Louis, Atlanta, Dallas, San Francisco, Los Angeles, Portland, Seattle, Montreal.



Compressors must give dependable service under gruelling conditions of rough terrain, mud, and constant exposure to the shock and vibration of drilling. Unless protected from shock and vibration, even the most rugged equipment is subject to excessive down-time and maintenance.

Ingersoll-Rand Company solves this problem in its rugged mobile "Quarrymaster." Each of the 10,000-pound compressor power plants is protected by four LORD Bonded-Rubber Mountings which prevent shock and vibration from interfering with efficient performance. They also prevent excessive twisting stresses in the power plant when the "Quarrymaster" is moving over uneven ground.

Whether your product is large or small, reduction of shock and vibration improves performance and accuracy . . . lowers maintenance costs . . . increases customer satisfaction. Investigate LORD Vibration Control Mountings and Bonded-Rubber Parts. Write to attention of Product and Sales Engineering Department.

LORD MANUFACTURING COMPANY, ERIE, PA.

Canadian Representative: Railway & Power Engineering Corporation, Ltd.

LORD BONDED RUBBER

Vibration-Control Mountings . . . Bonded-Rubber Parts

of the state, with **George Schmitt** and **George Evans** continuing to represent the company in Ohio. At the same time it was announced that **Benedict H. Meyer** has been appointed as representative for Zapon finishes in Iowa, Nebraska, Kansas and western Missouri.

♦

With twenty-five years' experience in the rubber business, **J. R. Keach** of Cleveland has been appointed general manager of Quaker Rubber Corp., Philadelphia, division of H. K. Porter Co. Inc., Pittsburgh. He has served as purchasing agent for the Ohio Rubber Co., Willoughby, O.; general plant manager for Firestone Industrial Products Co., Akron; and directional sales manager for the Hamilton Rubber Corp., Trenton, N. J.

♦

E. W. Connolly has been appointed sales manager of the Detroit district of Haynes Stellite Division, Union Carbide and Carbon Corp. Mr. Connolly has been associated with the division for eleven years as sales engineer in the Detroit district. He succeeds **R. D. Gunther,** who has retired.

♦

Chicago manufacturer of hydraulic and pneumatic production equipment, the Hannifin Corp. has announced the transfer of **George O. Hendee** to field service and sales engineer for New England, where **Elliott D. Thompson** of Boston has withdrawn as sales representative after more than twenty years with the firm. Mr. Hendee, for over four years Hannifin sales and service engineer in the Philadelphia area, will make his headquarters at Worcester, Mass. He is succeeded at Philadelphia by **Joseph O'Malley,** who will serve eastern Pennsylvania, southern New Jersey and Delaware.

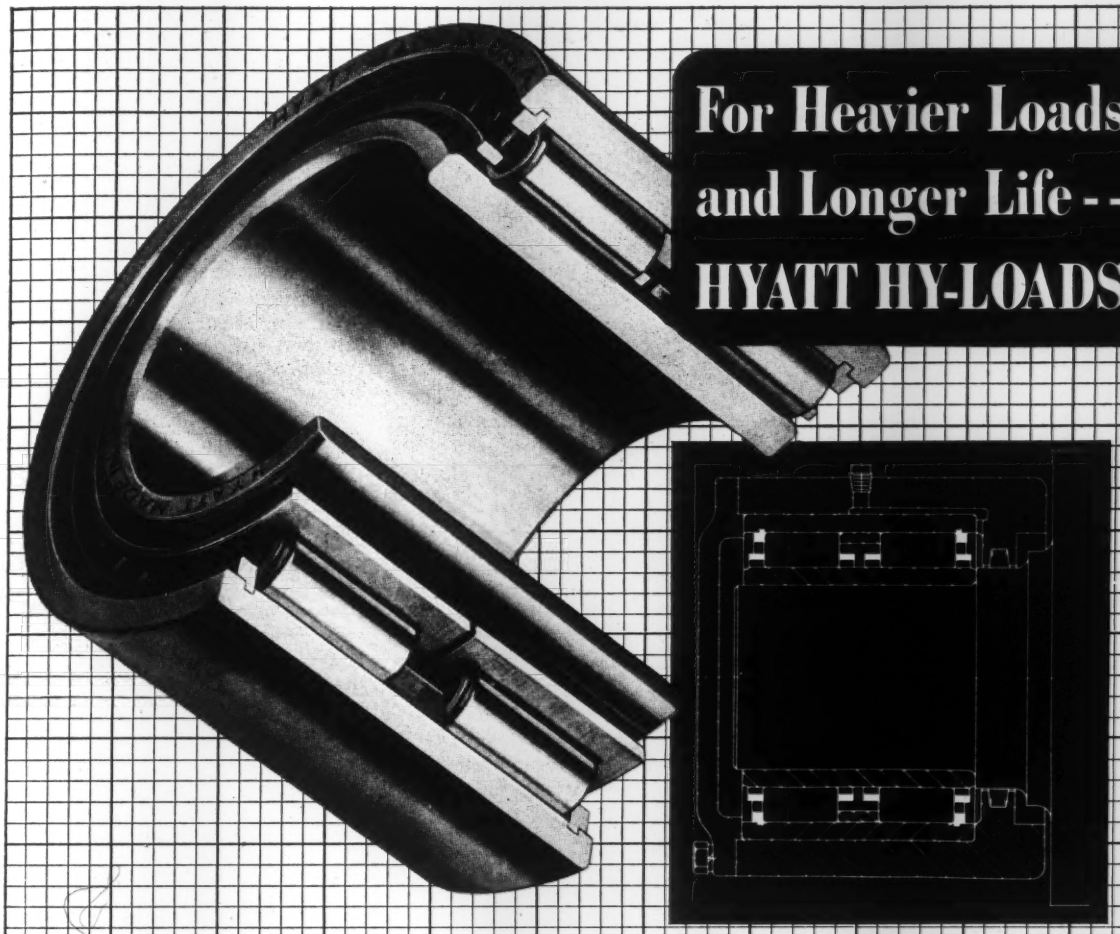
♦

Appointed to represent the company in territory including the entire state of Louisiana and parts of Florida, Mississippi and Alabama, **David L. Cushing** has joined the Republic Rubber Division of Lee Rubber & Tire Corp., Youngstown, as a field engineer.

♦

The board of directors of Automatic Steel Products Inc. of Canton, O., has elected **H. O. Holland** vice president in charge of sales of all divisions of the corporation. Mr. Holland has been director of sales since 1947 and was formerly vice president of the Kellogg division of American Brake Shoe Co. **John E. Carnahan** has been named vice president of the spun steel division of the corporation.

For Heavier Loads and Longer Life -- HYATT HY-LOADS



The cut-away Hyatt Hy-Load Roller Bearing illustrated is the A-TS Duplex Type, 6200 Series. Duplex type Hy-Load bearings are much wider and have much greater load capacity than other standard series bearings of the same bore size. Two roller-separator sets are assembled with the outer race as a unit. The roller sets are spaced by a steel ring centered in the bore of the outer race.

While Duplex Bearings can sustain extremely heavy loads, the allowable deflection of the shaft is usually a limiting factor. The extra capacity is therefore utilized, in most cases, to meet a greater life requirement.

The straight, separable inner race type of construction makes possible simple methods of assembling machine parts and most efficient lubrication sealing. It also allows axial expansion or slight oscillation without cramping the roller ends.

The separable parts are freely interchangeable as any inner race will fit any roller assembly of the same number. The inner race can be omitted and the rollers operated directly upon a hardened and ground shaft.

The drawing shown above is a conventional bearing arrangement for a long shaft or roll and illustrates the application of a 6200 series bearing. All Hyatt Hy-Load Bearings are shown in our new Hy-Load Bearing Catalog, number 547. Write for your copy today. Hyatt Bearings Division, General Motors Corporation, Harrison N. J.

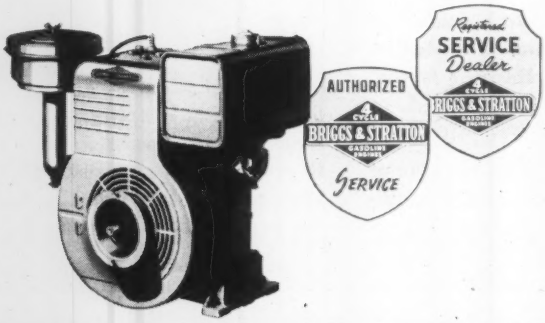
HYATT ROLLER BEARINGS

Through the Largest Network of its Kind in the World

There is a Briggs & Stratton service organization, factory-trained and supervised, near you — with a complete stock of genuine Briggs & Stratton parts for all models.

Briggs & Stratton service is world-wide — and constitutes the largest network of its kind. It is an important part of Briggs & Stratton's continuous effort to make its engines of greatest constant value to their users.

BRIGGS & STRATTON CORP., Milwaukee 1, Wisconsin, U.S.A.



In the automotive field Briggs & Stratton is the recognized leader and world's largest producer of locks, keys and related equipment.

SALES NOTES

EXTENSION of their Pacific Coast representation to the San Francisco area and northern California has been announced by **Linear Inc.**, Philadelphia manufacturers of hydraulic packings and precision molded rubber products. The **Haskel Engineering & Supply Co.**, Linear's exclusive West Coast representative, with main offices at Glendale, Calif., has opened a full service branch office at 1245 22nd St. in San Francisco under the management of E. C. Osborn. Complete stocks of specification or commercial O-rings, as well as many other types of mechanical packings such as asbestos, flax, duck and rubber, semimetallic, plastic and sheet will be avilable at the San Francisco office.

♦

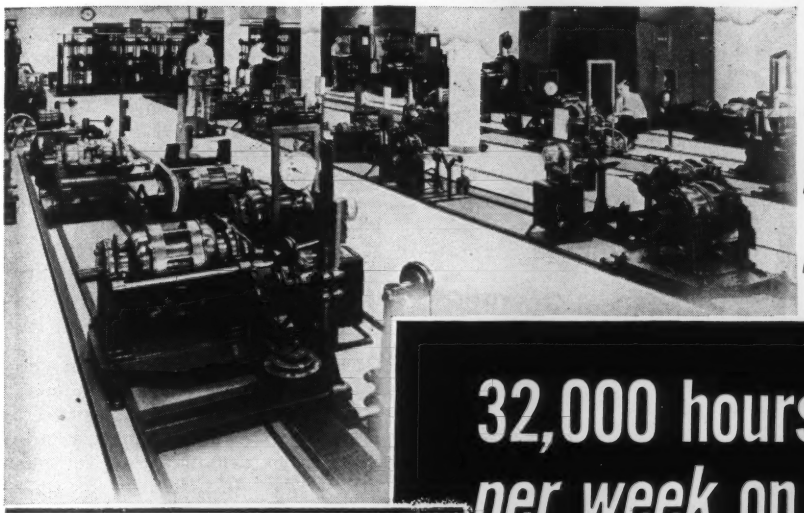
A new sales office was opened recently in Philadelphia by the **John S. Barnes Corp.**, Rockford, Ill. The new office will be operated by **Ernest A. Isberg and Co.**, Theatre Square, Swarthmore, Pa., and will serve the eastern Pennsylvania, southern New Jersey and Delaware areas.

♦

Ampco Metal Inc., Milwaukee, Wis., has announced that **Jaeger Welding Supply Inc.**, Springfield, Mass., and the **Texas Welding Supply Co.** of Dallas, Tex. have been appointed as franchised distributors in their respective areas. Each will handle the complete line consisting of various grades of Ampco-Trode, Sil-Trode and Phos-Trode electrodes.

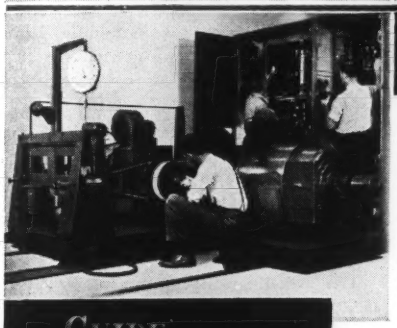
♦

In line with company policy to extend field contacts through trained representatives equipped to work closely with engineers and operating men on problems pertaining to flexible lines for all purposes, the **Chiksan Co.** of Brea, Calif., has appointed five new representatives. **Robert S. Hudgins Jr.**, 118 North Chase St., Charlotte, N. C., will cover North and South Carolina. The **Southern Corp.**, Charleston, S. C., will continue to represent Chiksan in the pulp and paper industry in these states. **R. S. Stover Co.**, Kresge Bldg., Marshalltown, Iowa, has been appointed to



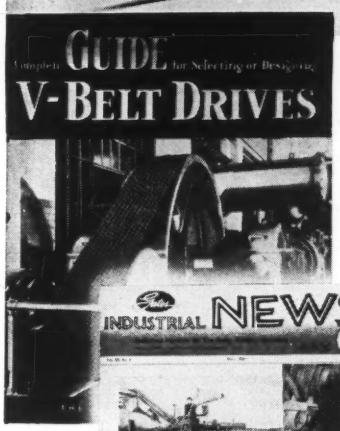
In the GATES Research Laboratories...

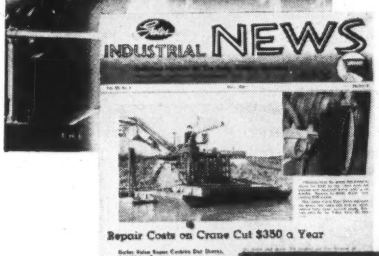
32,000 hours of testing *per week* on V-Belts alone!



In Gates V-Belt testing laboratories, which are by far the largest in the world, we run *32,000 hours of testing per week* on V-Belts alone. Thats' the equivalent of *5 years of continuous service* for one belt—*all in a single week!*

This intensely *Specialized* Research on V-Belts has naturally given Gates Engineers a knowledge of V-Belts and V-Belt drive design that is truly unusual—and it is this specialized knowledge that is embodied in the Gates "GUIDE for SELECTING or DESIGNING V-BELT DRIVES."





Take Advantage of this Great Store of V-Belt Knowledge—FREE!

The GUIDE is widely recognized as the most comprehensive and complete book of its kind—as well as the *easiest to use*—and it is *yours for the asking!*

Another Gates publication, the "INDUSTRIAL NEWS," published monthly, brings you the case histories of actual V-Belt Drives that have been designed and used by practical operating men or by plant engineers—reporting all that is new and progressive in V-Belt drive practice.

Whether you are a practical operating man or a design engineer, you can save much time, hard work and money by having these very useful Gates publications. They are entirely free—are sent to you without the slightest obligation. Just mail the coupon below—today!

Get these TIME, WORK and MONEY SAVING PUBLICATIONS FREE

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I would like you to send me without cost or obligation—

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Name of firm_____

City_____ State_____

ENG-307

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Gates VULCO ROPE DRIVES

Engineering Offices and Jobber Stocks IN ALL INDUSTRIAL CENTERS of the U. S. and 71 Foreign Countries

THE GATES RUBBER COMPANY

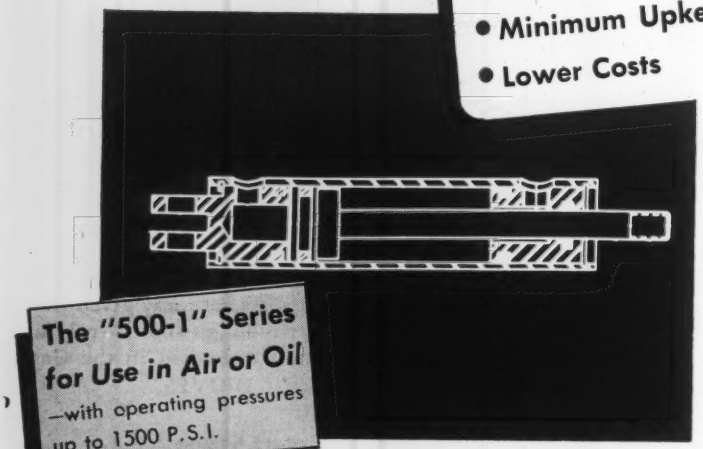
The World's Largest Makers of V-Belts

DENVER, U. S. A.

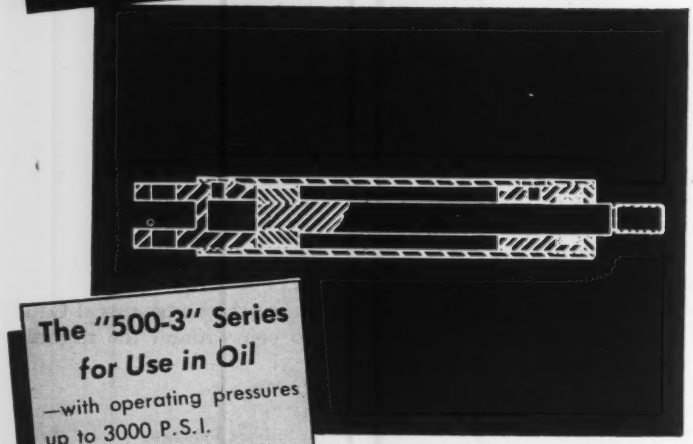
Specify Electrol HYDRAULIC CYLINDERS

Where Applications Call for

- Simpler Design
- Precise Construction
- Lighter Weight
- Economical Operation
- Better Service
- Longer Life
- Minimum Upkeep
- Lower Costs



The "500-1" Series for Use in Air or Oil
—with operating pressures up to 1500 P.S.I.



The "500-3" Series for Use in Oil
—with operating pressures up to 3000 P.S.I.

Special Cylinders

No matter how specialized your requirements may be—keep in mind that Electrol both designs and builds hydraulic cylinders of every type. Write for details, or let us have one of our engineers call at your convenience.

Electrol
INCORPORATED
KINGSTON, NEW YORK
CYLINDERS • SELECTOR VALVES • FOLLOW-UP VALVES
CHECK VALVES • RELIEF VALVES • HAND PUMPS
POWERPAKS • LANDING GEAR OLEOS • SOLENOID
VALVES • ON-OFF VALVES • SERVO CYLINDERS • TRANSFER
VALVES • CUT-OUT VALVES • SPEED CONTROL VALVES
FOR BETTER HYDRAULIC DEVICES

cover the states of Iowa and Nebraska, and the **Geo. S. Thomson Co. Inc.**, 915 South Central Ave., Phoenix, Ariz., will represent the Chiksan Co. in the state of Arizona. **Jos. W. Eshelman & Co. Inc.**, 2625 Sixth Ave., South, Birmingham 5, Ala., is the new representative covering Alabama and parts of Tennessee and Florida, and **Rittelmeyer & Co. Inc.**, 150 Nassau St., N. W., Atlanta 1, Ga., will cover the entire state of Georgia and a part of Florida.

♦

The Robt. W. Hoffman Co. of Chicago, distributors of Acme & Acro spot welders, has announced a recent reorganization and expansion program in the field of resistance welding. Robt. H. Hoffman and P. Arthur Green, formerly of Chicago Welder & Machine Co., Hinsdale, Ill., have consolidated their companies to offer a combined welding sales and service organization. The new showroom, welding laboratory and shop is located at 32-34 South Clinton St., Chicago, Ill. The firm will operate under the name of **Robt. W. Hoffman Co. Inc.**

♦

The Pacific Coast plant of **Parker-Rust Proof Co.**, manufacturers of Bonderite, Parco Compound, Parco Lubrite and other metal preparation and cleaning products, is now in operation at 3710 Fruitland Ave., Maywood, Calif. The new plant has been established to bring western industry the advantages and economies of quick supply and on-the-spot manufacture of these products. Many Parker products are now being manufactured at the Maywood plant, and more will be made there in the future.

♦

Atlas Chain and Manufacturing Co. has improved its representation in the North Jersey and Paterson-Passaic area by appointing **Bernstein Brothers Inc.**, 1-17 Keen St., Paterson, N. J., as distributors for Atlas roller chain.

♦

The **DeLaval Steam Turbine Co.** of Trenton, N. J., has announced the appointment of **Wilkinson & McClean Ltd.** of Calgary, Alberta, Canada as sales representative for that province. They will handle De Laval turbines, compressors, centrifugal pumps, speed reducers, couplings and IMO screw-type rotary pumps. Specialized engineering projects and marine equipment will continue to be handled by the De Laval organization in Trenton.

Cost Reduced 12%
Weight Reduced 5%
Dependability Assured with STEEL CASTINGS

This is an idler gear rocker arm. It is a part of a steam locomotive booster. Weight before redesign to a steel casting — 147 pounds.

Although the manufacturer was making the rocker arm as a weldment in his own plant, he was interested in discussing the advantages of a steel casting, which he would have to buy outside.

Consultation between the engineering department of the manufacturing company and the steel foundry engineer resulted in a 12% lower cost and 5% lower weight plus other important benefits. A marked increase in dependability was obtained through greater rigidity with toughness, and additional strength where needed — because of better distribution of the metal.

Conversion of some of *your* high-strength parts to steel castings may result in equal if not greater benefits than those shown here.

Also in new and redesigned parts, your steel foundry engineer may be able to suggest ways to cut costs and weight and obtain other substantial benefits . . . if you call him in for a discussion *while your product is still on the drawing board.*

This service is offered without cost or obligation. It makes available through your foundry engineer the full results of the development and research program carried on by Steel Founders' Society of America.

STEEL FOUNDERS' SOCIETY OF AMERICA

EXTRA STRENGTH

FOR SPECIFY GENUINE ALLEN ⬡ HEAD SCREWS

The Allen Pressur-Forming Method is now used to produce nearly all standard Allen screws. Instead of weakening the metal by cutting the steel fibres, it *compresses* them for *extra* toughness.

IF YOU BUY FOR REPLACEMENT . . .

you'll buy less often if you are sure to get the extra strength Allen builds into precision fastenings.

IF YOU ARE DESIGNING OR IMPROVING A PRODUCT . . .

Allen technical development (available through Allen distributors or direct from the factory) leads the field. We work constantly with engineers of leading manufacturers toward the solution of problems involving fastenings and we invite your inquiry.

ASK HOE . . . There's no room for failure in the giant high speed presses that turn out America's newspapers — but there's room for thousands of space-saving Allen ⬡ Head screws in every Hoe press. This leading manufacturer standardizes on Allen⬡Head screws for dependability.

SOLD ONLY THROUGH LEADING DISTRIBUTORS

Write the factory direct for technical information and descriptive literature.

WARNING
Allen-Type screws aren't necessarily Allen-Made. Get genuine "Pressur-Formed" Allen⬡Head screws in this black and silver box.

ALLEN ALLEN HEAD

MANUFACTURING COMPANY

Hartford 2, Connecticut, U. S. A.

NEW YORK, CLEVELAND, DETROIT, CHICAGO, LOS ANGELES

MEETINGS AND EXPOSITIONS

Aug. 7-19—

International Trade Fair to be held in the Coliseum, Arena, International Amphitheater and Navy Pier, Chicago, Ill. Additional information may be obtained from the First United States International Trade Fair, 10316 Merchandise Mart, Chicago 54, Ill.

Aug. 14-16—

Society of Automotive Engineers. West coast meeting to be held at the Biltmore Hotel, Los Angeles, Calif. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Sept. 11-14—

Society of Automotive Engineers. Tractor meeting to be held at the Schroeder Hotel, Milwaukee, Wis. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Sept. 18-22—

American Society of Mechanical Engineers. Instruments and Regulators Division Conference to be held at the Municipal Auditorium, Buffalo, N. Y. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Sept. 18-22—

Instrument Society of America. The fifth national conference and exhibit to be held in the Memorial Auditorium, Buffalo, N. Y. Richard Rimbach, 1117 Wolfendale St., Pittsburgh 12, Pa., is executive secretary.

Sept. 19-21—

American Society of Mechanical Engineers. Fall meeting to be held at the Hotel Sheraton, Worcester, Mass. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

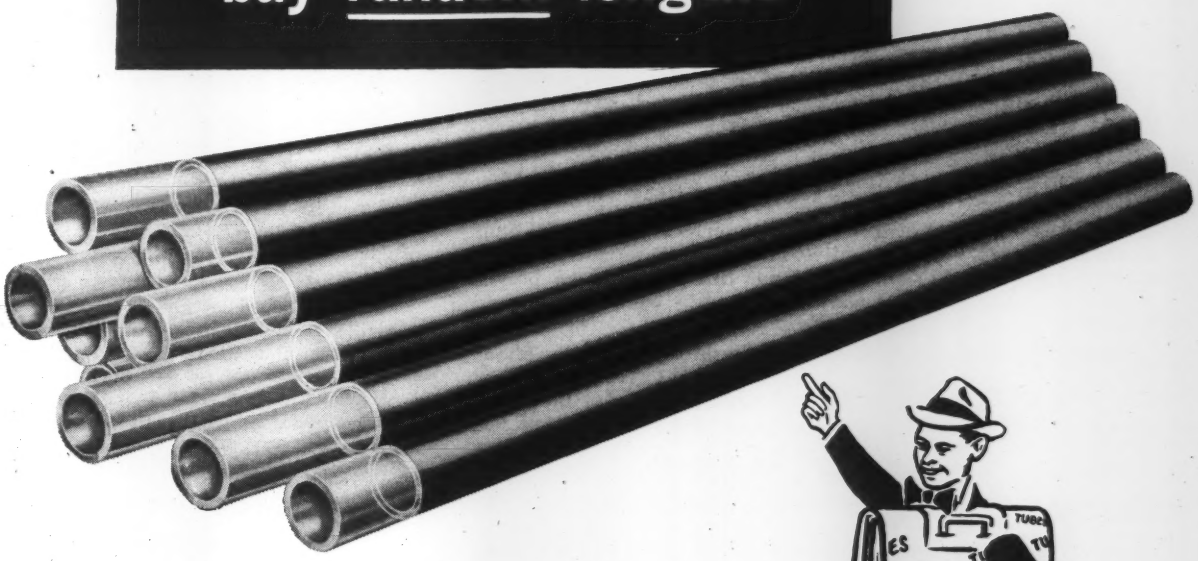
Sept. 25-27—

American Society of Mechanical Engineers. Petroleum mechanical engineering conference to be held at the Hotel Roosevelt, New Orleans, La. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Sept. 26-29—

Association of Iron and Steel Engineers. Annual iron and steel exposition to be held at the Cleveland Public Auditorium, Cleveland, Ohio. Additional information may be obtained from Society headquarters,

. . . might pay you to buy random lengths



Ask a B&W Tube Representative . . . he knows

Perhaps you are *unnecessarily* exacting in the specification of length on your mechanical tubing orders. Is the added expense justified by end-use and manufacturing methods?

FOR EXAMPLE:

If your product and production methods permit, you can save up to 5% by ordering select random lengths of B&W seamless or welded mechanical tubing . . . 3½% on random multiple lengths. B&W Tube representatives, factory trained in "tube-technics", can often help you uncover economies in the purchase and application of B&W mechanical tubing.

B&W SEAMLESS and WELDED TUBES

THE BABCOCK & WILCOX TUBE COMPANY

General Offices: Beaver Falls, Pa.

Plants: Alliance, Ohio, and Beaver Falls, Pa.

Sales Offices: Alliance, Ohio • Beaver Falls, Pa. • Boston 16, Mass. • Chicago 3, Ill. • Cleveland 14, Ohio • Denver 2, Colorado • Detroit 26, Mich. • Houston 2, Texas • Los Angeles 15, Calif. • New York 16, N. Y. • Philadelphia 2, Pa. • St. Louis 1, Mo. • San Francisco 3, Calif. • Syracuse 2, N. Y. • Toronto, Ontario • Tulsa 3, Okla. • Vancouver, British Columbia

TA-1565M

The doctor held his breath. An iron lung had stopped. For four years it had kept a little girl alive. Now . . . ? Feverishly, hospital attendants readied a temporary lung. But it would be only a matter of hours before substitute lung and young life would snuff out together.

Five hundred miles away in the Winsmith plant, the phone jangled with an urgent question. Could they? They'd *have* to! The clock on the wall said three. Experts cooly raced the hands of the clock and at four, the gravely needed speed reducer had been assembled and flown on its way.

That night, the little patient . . . *everyone* breathed easier. Winsmith had won a race against time.

Unusual? Not altogether, especially when an equipment manufacturer has the foresight to install a *standard* reducer. Yes, meeting and beating tough challenges is an old Winsmith habit, thanks to standardization and management's responsiveness to customers' problems.

Your speed reducer problems may not be a matter of life or death, but they're often a matter of profit or loss. Winsmith can help you solve these problems to your best interests . . . time-wise and cost-wise. One of the reasons . . . industry's widest range of standardized types and sizes up to 85 H.P. Another . . . the large inventory of standard parts ready for quick assembly. Together, they enable the Winsmith field engineer in your territory to prescribe to your needs right out of stock from one of these 3 basic types:

FREE CATALOG
Handbook No. 148 with complete engineering data. Write.

1010 Empire Bldg., Pittsburgh 22, Pa.

Oct. 12-13—

Gray Iron Founders' Society. The 22nd annual meeting to be held at the Netherland-Plaza Hotel, Cincinnati, Ohio. R. L. Collier, 210 National City Bldg., Cleveland 14, Ohio, is executive vice president.

Oct. 16-18—

Society of Automotive Engineers. Transportation meeting to be held at the Statler Hotel, New York, N. Y. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

Oct. 18-19—

National Conference on Industrial Hydraulics. Sixth annual meeting to be held at the Sherman Hotel, Chicago, Ill., under the sponsorship of the Illinois Institute of Technology and local sections of ASCE, ASME, SAE, Western Society of Engineers, ASLE, AIChE and IAS. Frank W. Edwards, Illinois Institute of Technology, 3300 South Federal St., Chicago 16, Ill., is conference secretary.

Oct. 18-20—

Society of the Plastics Industry. Annual national conference to be held at New Ocean House, Swampscott, Mass. William T. Cruse, 295 Madison Ave., New York 17, N. Y., is executive vice president.

Oct. 23-25—

American Gear Manufacturers Association. Semi-annual meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Newbold C. Goin, Empire Bldg., Pittsburgh, Pa., is executive secretary.

Oct. 23-27—

American Institute of Electrical Engineers. Fall meeting to be held at the Skirvin Hotel, Oklahoma City, Oklahoma. H. H. Henline, 33 West 39th St., New York 18, N. Y., is secretary.

Oct. 23-27—

National Metal Congress and Exposition. The 32nd annual metal congress and exposition to be held at the Chicago International Amphitheatre, Chicago, Ill. W. H. Eisenman, 7301 Euclid Ave., Cleveland, Ohio, is managing director of the exposition.

Nov. 1-3—

American Society of Body Engineers, Inc. Fifth annual technical convention to be held at the Rackham Memorial Bldg., Detroit, Mich. Additional information may be obtained from Society headquarters, 100 Farnsworth Ave., Detroit 2, Mich.

New Property Chart of ALSIMAG®

TECHNICAL CERAMIC COMPOSITIONS

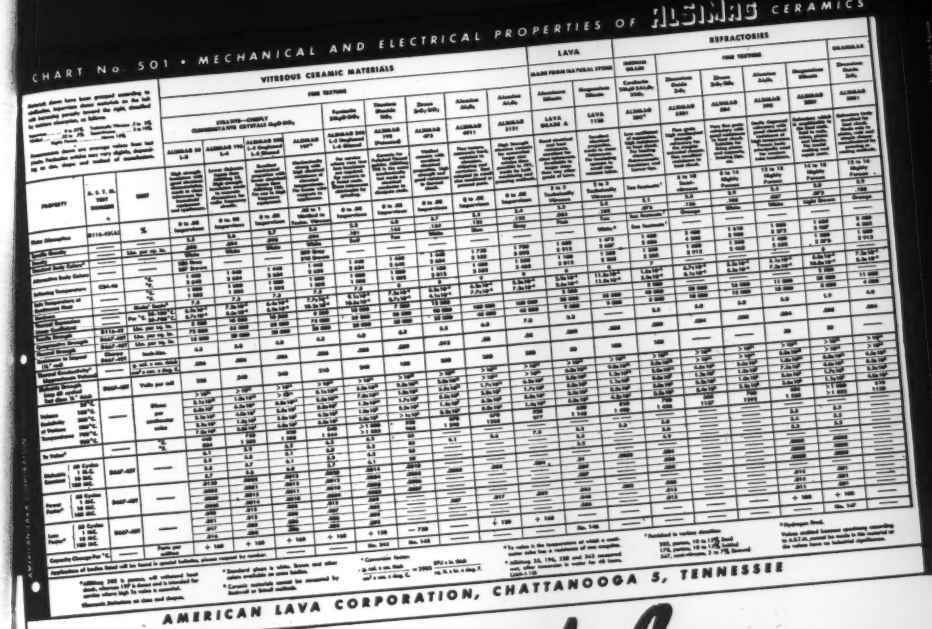
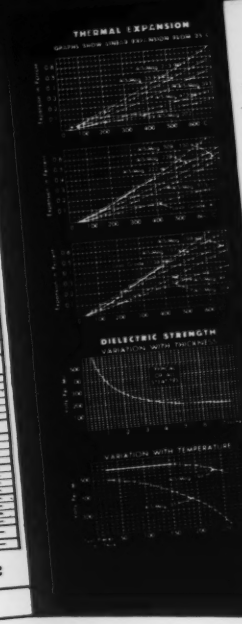


CHART No. 501 • MECHANICAL AND ELECTRICAL PROPERTIES OF ALSIMAG CERAMICS

VITREOUS CERAMIC MATERIALS

LAVA

REFRACTORIES


THERMAL EXPANSION

DIELECTRIC STRENGTH

VARIATION WITH TEMPERATURE

SELECTION CHARTS

AMERICAN LAVA CORPORATION, CHATTANOOGA 5, TENNESSEE

sent free on request

American Lava Corporation, Chattanooga 5, Tennessee, has issued a new chart giving the mechanical and electrical properties of AlSiMag custom-made technical ceramics.

WHAT ALSIMAG IS

AlSiMag is the trade name of a large family of technical ceramic compositions. These compositions have different physical, electrical, mechanical and chemical characteristics. AlSiMag ceramics are custom-made to specifications.

WHAT THE CHART TELLS

The chart covers seventeen of the more frequently used AlSiMag compositions and is the most complete chart yet issued in this field. A new feature is a selection chart which simplifies and speeds the selection of the most useful composition for the individual requirement. This selection chart indicates lower cost materials in BOLD FACE. This helps the product engineer to design for utmost economy.

Some properties, such as thermal expansion, dielectric strength, in relation to thickness and temperature are presented in graphic form.

ALSIMAG COMPOSITIONS NOT ON CHART

Many special AlSiMag compositions have been developed to meet specific conditions. These are too numerous to chart. If chart indicates general characteristics of value, modifications to suit your special application may be available.

WHO NEEDS THE CHART

Designing engineers, production technicians or purchasing agents will find chart helpful in their search for materials for unusual applications.

HOW TO GET THE CHART

The AlSiMag Property Chart is sent free on request. Request as many copies as you need to cover your organization.

WHERE ALSIMAG IS USED

AlSiMag custom-made technical ceramic parts are extensively used as:

Insulators for the electronic field • Insulators for electric appliances and other electrical applications • Thread Guides for textiles, wires, paper twine, etc. • Extrusion dies for such products as pencil leads, battery carbons, soft wires, explosives, etc. • Gas burner tips • Controlled atmosphere welding tips • Oil burner ignition insulators • Ceramics for hermetic seals • Metal-ceramic combinations • Air-acid jet nozzle inserts • Polishing heads for delicate final polishing operations • Cores and inserts for precision castings • Strainer cores for metal foundries • Cut-off cores for metal foundries • Refractory pins and plates in small sizes and special shapes • Work holders for electronic heating devices • As a replacement for parts made of plastic, wood or machined metal wherever a wear resistant part is required • In short, wherever electricity, heat, chemical or certain abrasive or friction conditions must be controlled.

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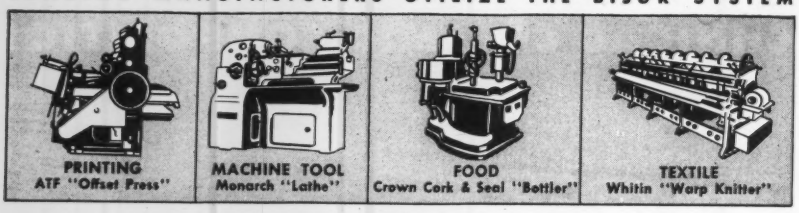
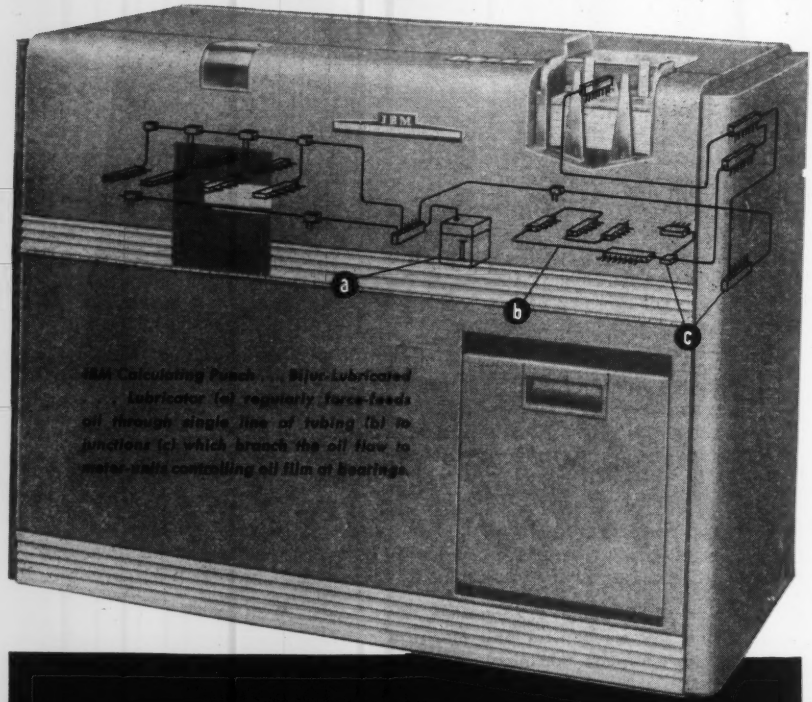
LEADING MANUFACTURERS UTILIZE THE BIJUR SYSTEM



built-in protection



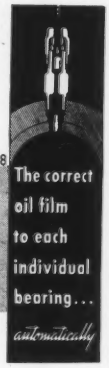
IBM Calculating Punch . . . Bijur-Lubricated . . . Lubricator (a) regularly force-feeds oil through single line of tubing (b) to junctions (c) which branch the oil flow to meter-units controlling oil film at bearings.

over 100 bearings lubricated...automatically

All 109 bearings of this machine must be oiled at once *during operation* to maintain continuous production. The Bijur system does the job by connecting all bearings to one lubricator, driven by the machine.

No sign of a system is seen outside the machine as it is completely built in at time of manufacture. This makes for a compact internal design and a clean-lined exterior. All bearings are oiled at once, yet each one receives individual attention from Bijur, the system with positive Meter-Unit control of oil flow *at the bearings.*

For further details write for "The ABC of Modern Lubrication."



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BIJUR
LUBRICATING CORPORATION
LONG ISLAND CITY 1, NEW YORK

DESIGN ABSTRACTS

Air Line Engineering

A specific definition of air line engineering is rather difficult to offer since it encompasses so many and such varied operations. The following will serve as a starting point: Air line engineering is the scientific utilization of available aircraft and other equipment in filling the needs of an individual airline. Airline engineering can be divided into three primary classifications: aircraft engineering, flight engineering, and aircraft maintenance engineering.

AIRCRAFT ENGINEERING: Consists of a study of the performance characteristics of available and projected transport aircraft as applied to the particular route system, with an eye toward determining the most suitable type of aircraft to obtain maximum utilization, highest possible overall traffic load factor and lowest operating cost. After the selection of an aircraft has been made, a detailed study of its component systems follows to determine their suitability for maximum reliability and ease of maintenance, based on past operating experience.

FLIGHT ENGINEERING: Consists of a study of the aircraft operating manuals offered by the manufacturer and their modification to conform with the individual air line system regulations. At the same time a determination is made as to the best cruising power and altitude for any given route segment to meet schedules and cost considerations.

AIRCRAFT MAINTENANCE ENGINEERING: Covers the problems of absorption of a new type of aircraft into a particular air line system. In order to do this it is necessary that many modifications be effected during the first stages of operation. These modifications stem from the need for greater efficiency, changes in government regulations, or by pressure of competition for the attraction of passengers.

Qualifications

Qualifications of the air line engineer are:

1. A well grounded knowledge of the fundamental engineering subjects of mathematics, physics, drafting and applied mechanics. Attention should be given also to fundamental courses

XUM

it's easy

if you know how

As versatile as stainless is, you may find it difficult to apply stainless steel to your job unless you know which stainless analysis you need. That's why Crucible, pioneers in the development of this specialty steel, makes freely available to you an alert metallurgical staff. These engineers and metallurgists can show you which stainless analysis is best suited for your application. And how Crucible Stainless can be applied for maximum satisfaction.

Take full advantage of Crucible's half century of specialty steel leadership — leadership based on a keen devotion to detail . . . on every Industry-posed problem . . . whether the order is in tons or pounds. When you think of stainless . . . call in Crucible. CRUCIBLE STEEL COMPANY OF AMERICA, Chrysler Building, New York 17, N. Y.

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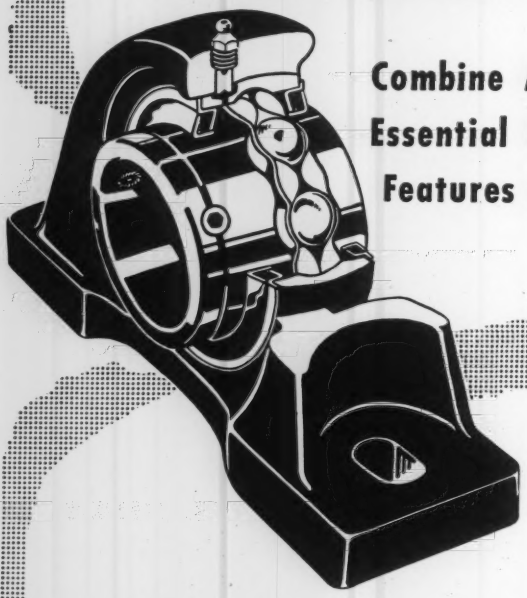
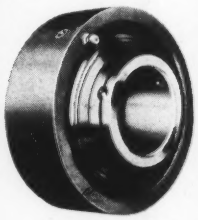
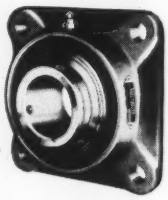
BALL BEARING UNITS



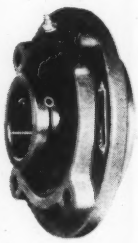
Combine All Essential Bearing Features In One Unit!



Cartridge Units



Flange Units



Flange Cartridge Units

SEALMASTER Ball Bearing Units have features that contribute to the good performance and long life of any industrial product on which they are used. Textile equipment, oil field machinery, farm implements and air conditioning equipment are only a few of the many widely varied fields in which manufacturers have standardized on SEALMASTER units.

To be sure of the best possible bearing performance, consider SEALMASTER units when designing, or redesigning products. Write us for a copy of SEALMASTER Catalog No. 845.

Here is the exclusive combination of features found only in SEALMASTER Ball Bearing Units:

- **Permanently Sealed** — Felt-lined, steel flinger, rotating in labyrinth, keeps dirt out and retains lubricant . . . prolongs bearing life.
- **Self-Aligning** — Bearing unit with seals independent of housing can align itself in any direction without seal distortion.
- **Pre-Lubricated** — Bearings are factory lubricated and are shipped ready for immediate use.
- **No Housing Wear** — Patented locking pin prevents rotation of outer race in housing and positions bearing for relubrication . . . no housing wear means quiet operation.

BEARING DIVISION

STEPHEN S-A ADAMSON MFG. CO.

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in electricity and to sufficient work in advanced drafting.

2. A thorough knowledge of English, economics, industrial management and business administration is now considered necessary in the modern concept of engineering education.

3. A fundamental training in the practical process of sheet metal fabrication, machine shop practices and other mechanical skills from the point of view of production.

4. The need of the ability which we might title "engineering depth perception". This particular ability consists of a thorough grounding in the fundamentals of engineering and a deep and sincere interest in this type of work, as well as an open eye toward the limitations of production methods.

5. A better-than-average level of intelligence and a natural degree of versatility. The air line engineer is not only faced with the problems of aircraft or aircraft maintenance engineering, but also with the problems relating to airport design, flight traffic control, comparative evaluations of equipment and the design of auxiliary service devices required in the ground handling and loading of aircraft and in the maintenance shops.

Need for Engineers

Need for the aeronautical engineer in the air line industry is greater than has ever been properly met. The need is evidenced by the economies that engineering has effected in the past, and it is difficult to see how the air lines cannot afford more and better engineering talent to effect even greater economies in the future. *From a paper by Edward H. Barker, assistant dean, Parks College of Aeronautical Technology of St. Louis University, at SAE annual meeting, Jan., 1950, in Detroit.*

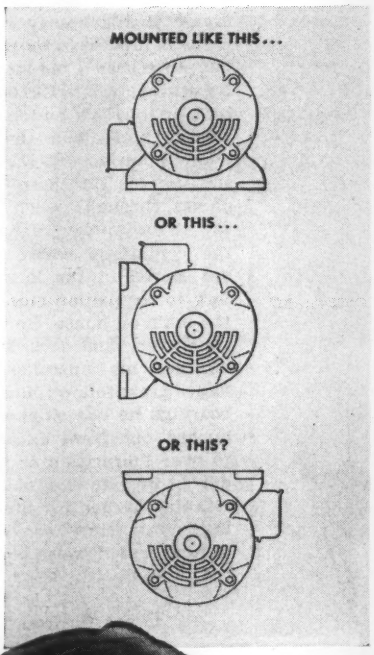
Proper Lubrication Practice

THERE are many ways to reduce maintenance costs. The old adage, "a stitch in time saves nine," seems particularly apt in increasing the life of machines through intelligent use of lubricants.

The main road to preventive maintenance through lubrication starts at the machine design stage. No doubt you have noticed machines with what seems to be a "tacked on" system of lubrication. In some cases the lubrication is an after thought, sometimes a long time after. When the pipe fitters, the electricians or maintenance crew finish their respective jobs in installation, the lubrication system or devices for applying the lubricant

YOU CAN BE **SURE**.. IF IT'S
Westinghouse

Is your motor *really* dripproof?



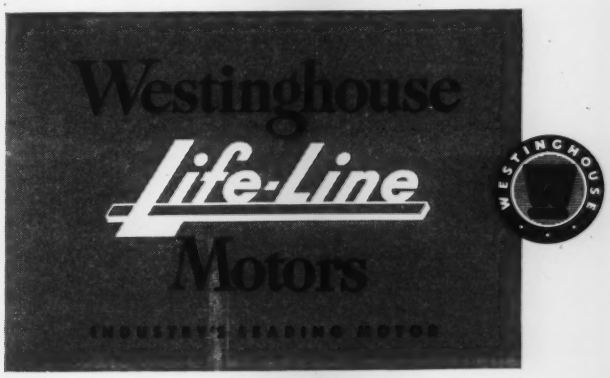
. . . AT ANY ANGLE OF MOUNTING— Do you see a ventilating opening in the frame? Then the motor is probably not dripproof when side wall or ceiling mounted. You need a special motor for these mounting positions . . . with special protection over frame openings. But . . .

You can save the costs of specials with Life-Line motors. Solid frame construction eliminates frame openings. All ventilation is accomplished through the end brackets. Simply rotate the 4-position brackets and the Life-Line is dripproof—for side wall or ceiling mounting.

Dripproof conduit box can be rotated also. Plenty of knockouts give maximum possible connections. And, because a Life-Line motor needs no further lubrication, you can crowd it into cramped quarters . . . mount it in "hard-to-get-at" locations—and forget it. Thus, you get maximum mounting flexibility for difficult mounting problems.

It costs no more to get these "special" features in a standard motor. Simply specify Life-Line as the drive for your equipment. Ask your local Westinghouse representative for details and a copy of booklet B-3842, or write Westinghouse Electric Corporation, P. O. Box 868, Pittsburgh 30, Pennsylvania. J-21583-A

If you figure LIFE-COSTS . . . you'll figure LIFE-LINE



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Over 17,000 copies of this famous 640-page Struthers-Dunn RELAY ENGINEERING HANDBOOK are now in use throughout the world! Replete with more than 1100 illustrations it is an ideal reference manual for everyone concerned with relay or timer application, use or maintenance. . . . *Price $3 per copy.*

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Fit the relay to the job—
NOT the job to the relay!

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BALTIMORE • BOSTON • BUFFALO • CHARLOTTE • CHICAGO • CINCINNATI
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are surrounded or buried by a number or pipes or conduits. It is almost impossible to clean out the sump without removing all or a portion of the piping, wiring or auxiliary equipment.

These machines come equipped with a variety of oil cups and grease fittings, in different sizes and locations, and with recommendation for trade name (or equivalent) lubricants. Often the design of these machines requires several lubricants to function efficiently.

The next link in this chain of unsystematized lubrication is the chap who applies the lubricants. Usually he is a fellow who has just graduated from a janitor's job, with a lift to his ego (he is now a lubrication engineer) accompanied by one step up in classification and a few cent's increase in his hourly rate.

He is broken in by the fellow (who has now been graduated to the next higher paid classification) who gives him an oil can and grease gun and follows him about the assigned territory pointing out the machines and the devices on these expensive machines through which the lubricants are to be applied. He is also shown the containers where the lubricants are stored in the local sections and how to requisition more of them from the main oil house. He is now ready to take over—and does. Like the doctor burying his mistakes, the maintenance crew follows him to replace the bearings he has skipped or that have burned out from excessive heat due to over lubrication and in some cases due to introduction of the wrong lubricant. Charges for this work, including down time or production loss during repair, must be placed against the manufactured product.

Great Savings Possible

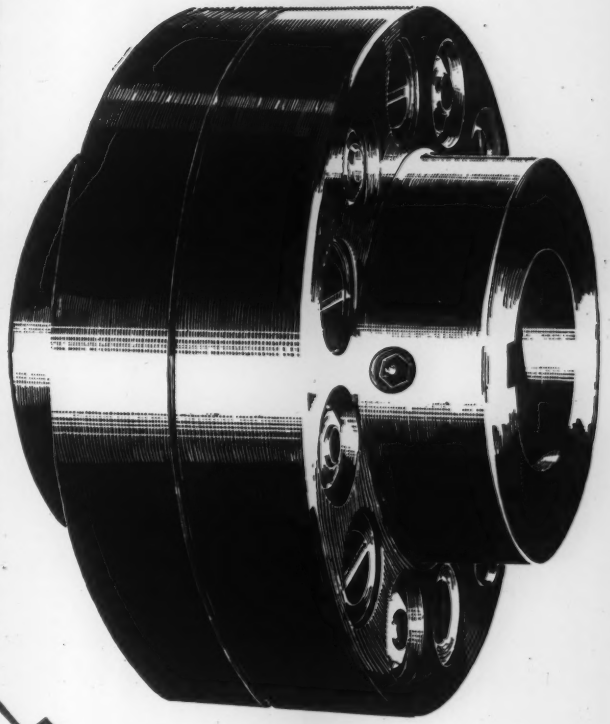
In many industrial plants an estimated mortality of bearings is expected. Maintenance costs and machine depreciation are set up from past experience. It seems apparent that here is an almost virgin field for savings. The fact that these savings are not clean cut and often are intangible makes it difficult to sell management on the importance of correct lubrication. This includes proper selection of both lubricants and lubricating devices under the supervision of one who is conscientious, has knowledge of the consequences of faulty lubrication and is able to help set up correct schedules for applying them and in the right amounts.

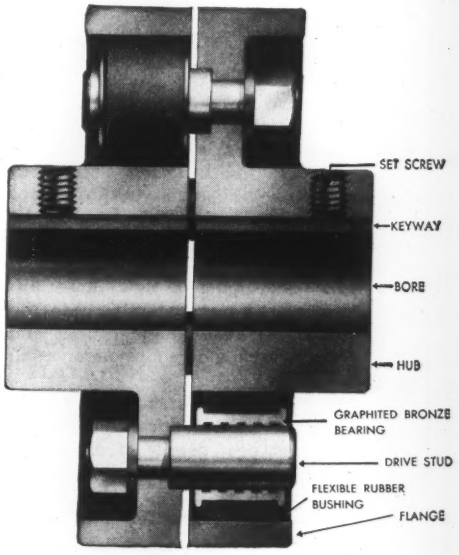
The ultimate goal includes: longer machine life, appreciable reduction in down time, more efficient operation, increased production at lower unit

How 30 years

OF SATISFACTORY PERFORMANCE HAVE MADE

AJAX





AMERICA'S STANDARD RUBBER BUSHED FLEXIBLE COUPLING

● Successful installations on a wide range of direct-connected machines have proved the design, performance and economy of Ajax Rubber Bushed Flexible Couplings.

Design engineers are using them as standard equipment for all types of applications including those requiring high operating speeds, increasing loads and space limitations.

They provide positive resilient drive, quiet operation and free end float. They require no lubrication and operate successfully in abrasive-laden air. They can be installed vertically or horizontally.

Standard Ajax Couplings are made of forged steel or cast semi-steel. Aluminum, bronze and other metals available for special applications. Made in a wide range of sizes and capacities . . . write for catalog.

AJAX FLEXIBLE COUPLING CO. INC.
WESTFIELD, N.Y.

Avoid . .
MACHINE "DOWN-TIME" CAUSED BY SOLENOID VALVE FAILURE
THE Bellows ELECTROAIRE* VALVE IS GUARANTEED AGAINST SOLENOID BURNOUT

COMPACT

The ¼" port size ELECTROAIRE VALVE can be hidden behind a package of "kingsize" cigarettes.

FAST

Speeds far beyond the speed ability of even short stroke present day air cylinders.

AMPLE POWER

Operates efficiently on any air pressure from 5 psi to 150 psi.

SAFE

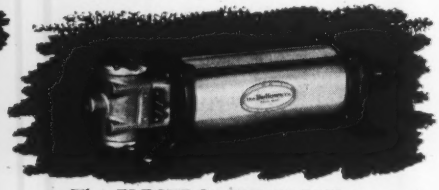
Low voltage assures safe operation even though flooded with coolant or piled with chips.

GUARANTEED

The ELECTROAIRE VALVE is unconditionally guaranteed against solenoid burnout.

With Bellows ELECTROAIRE VALVES in your pneumatic circuits you can forget worries about solenoid burnouts. The Bellows ELECTROAIRE VALVE offers safe, positive electrical control without the hazards of electrical failure. It is solenoid-controlled, *but air-powered.* It combines in one complete "packaged" air-control unit a four-way directional slide valve, solenoid controls, and piston speed regulators.

Its compactness, and easy adaptability to the most complex air and electrical circuits makes initial design for lower cost infinitely more practical. Its freedom from electrical hazards assures minimum "down-time"... minimum maintenance.



The ELECTROAIRE VALVE is standard equipment on all Bellows "Controlled-Air-Power" Devices arranged for electrical control. It is adapted for direct connection to or remote control of any air cylinder of suitable capacity.

Write for Your Free Copy of Bulletin AU-250 . . . specifications, wiring diagrams, dimensions of Bellows Electroaire Valves. Address: Dept. MD 750.

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The Bellows Co.
AKRON, OHIO

cost, and reduction in the number of machine repairmen needed. These savings go directly into the profit column of the company. *From remarks by George L. Sumner, lubrication engineer, Westinghouse Electric Corp., in a panel discussion at the First Plant Maintenance Conference jointly sponsored by SAM and ASME, Jan.* 1950, *in Cleveland.*

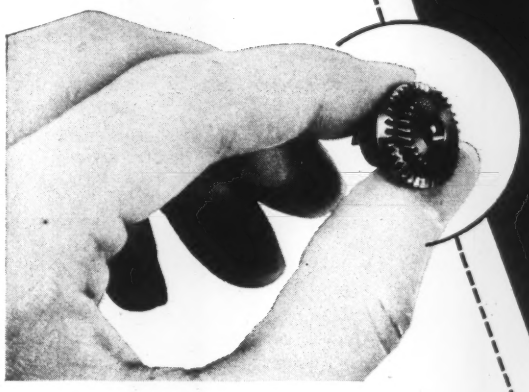
Aircraft Engine Design

THE feature of *aircraft* engines which sets them apart from others is the need for absolute dependability in a machine that is just as powerful and just as light as its size allows. We think of automobiles as being almost perfectly reliable, and they are, but they operate for most of the time at a mere fraction of full power. Furthermore, all large aircraft engines are supercharged, so that full power for them means harder work or higher loads and stresses than for an automobile engine.

Mechanically, the detail parts of aircraft engines are much like those of automotive engines, but fancier and more expensive alloys are freely used for nearly all parts. Stresses are higher; hence, deflections are larger and vibration is more troublesome. Fits and limits are much like automotive practice and in some cases clearances are more generous since noise is not such a consideration. Inspection is extremely rigid, and this applies right from the raw materials on down through all the steps of manufacturing and assembling. There is more machine work on the parts. Many parts are finished all over to reduce weight or to gain strength by having a smooth surface wherever high stress is to be carried.

There is little distinction between commercial and military engines. All the large engines used by the commercial airlines were designed originally for military use. The cost of developing a new large airplane engine is huge, running into the millions of dollars, and cannot and will not be shouldered by commercial operators.

From the nature and use of aircraft engines, their building entails the highest degree of effort in careful mechanical design. In other lines of machine design there is less urgency to obtain the very maximum of performance; new models can wait until the old tools are worn out or until improvements are needed to stimulate sales. The cost of producing aircraft engines is high because of the small quantities built and in the case of the larger sizes—1000 to 5000 hp—because of the enormous



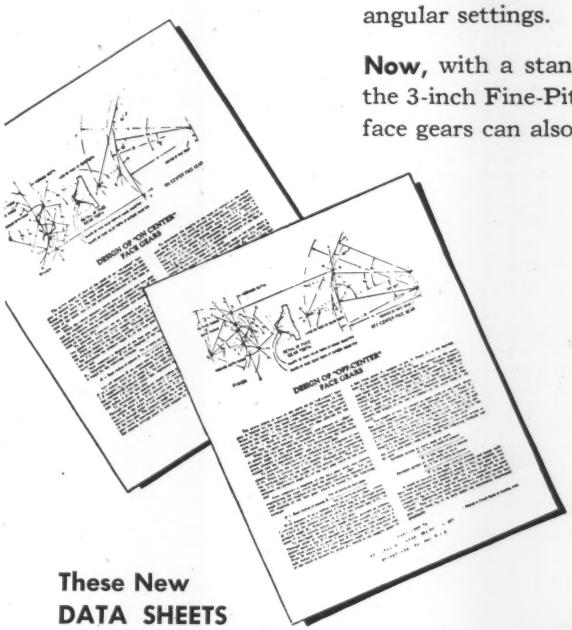
Involute Gear-Tooth Advantages in Angular Drives

A Face Gear with spur or helical teeth, driving or being driven by a gear or pinion, offers decided advantage for transmitting motion through shafts at 90 degrees or other angular settings.

Now, with a standardized FACE GEAR FIXTURE used on the 3-inch Fine-Pitch Gear Shaper, the following advantages of face gears can also be utilized for small-size units:

- **More compact design**
- **Lack of end thrust**
- **Flexibility in pinion positioning**
- **Sliding disengagement**
- **Ease of assembly**
- **Ratios from 3 to 1 and up**
- **Capacity for high speed**
- **Opportunity for reversal of rotation**
- **Ease and economy of manufacture**

A special bulletin on the principles and practical applications of face gears is available on request. Write the nearest Fellows office.



These New DATA SHEETS

explain a simplified procedure for determining the active face width for "on-center" and "off-center" face gears. Copies available.

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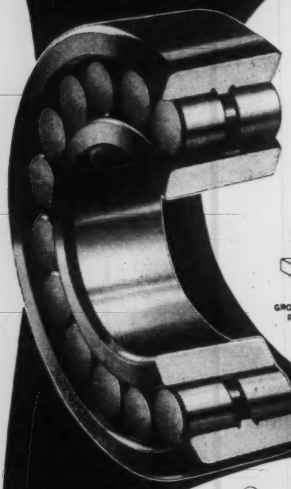
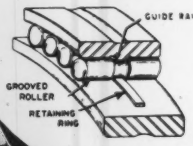
a
NEW PRINCIPLE
in
roller bearing design

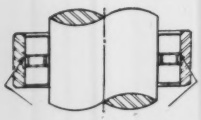




Rollers cannot skew. A guide rail integral with the outer race maintains alignment of the grooved rollers.

The NEW McGILL® GUIDEROL roller bearing

The Bearing with the Self-Aligning Rollers — a simplified roller bearing construction that automatically corrects roller skewing — without space wasting cages! Perfect roller alignment with resulting superior performance is maintained by this new principle of guided rollers. Even with reduced internal radial looseness a GUIDEROL bearing overcomes any tendency to bind under limited misalignment. Produced in needle and cylindrical type roller bearings. Write for information packed Bulletin today.

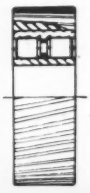


UNIVERSAL MOUNTING

Universal Mounting — from horizontal to vertical is possible. No cocking of rollers on vertical shafts because the grooved rollers are located by guide rail.

OPEN ENDS — SELF CLEANING

Open Ends — Self cleaning in applications permitting free circulation of oil or grease. Open faces set up flushing action inside the bearing.



GREATER ANGULAR RIGIDITY

Greater Angular Rigidity is provided by longer rollers that permit race and roller contact virtually to the outside edges of the races.

Write for your copy of the new GUIDEROL Bearing Bulletin No. GR-50. Address: McGill Manufacturing Co., Inc., Bearing Division, 200 N. Lafayette Street, Valparaiso, Indiana.

sums that must be spent in development and testing before a new model can be put out for general use.

It should be understood that development of a successful engine depends on a long course of experimental work in addition to the type of designing that is done with a pencil. Although no effort or cost is spared to make the most of theory and calculation, purely theoretical methods have been found inadequate for many purposes. For instance, many of the stresses and strains in engine parts are caused by vibration which is difficult to predict by theoretical studies. Then, even if the load on a part can be closely estimated, the shape of the part is generally so complex that it resembles nothing in any textbook on elastic theory and defies analysis of the stress distribution. This is not to belittle the value of theoretical consideration, since testing is costly and the closer one can come to the final workable design by calculation, the better off he is.

Long Test Period

We try to avoid excessive stresses and stress concentration. We try to anticipate and avoid vibration troubles. But after a new model engine is running, following perhaps a year of design work, there is a period of some five to eight years of test running and experimental development before the engine is ready for service. In other words, engine parts must be subjected to prolonged endurance tests under actual operating conditions to determine their ability to support operating loads. As experience on parts is accumulated, design factors are established and new engine designs benefit from past experience.

In order to keep weight down, parts that are hardly strong enough from the designer's standpoint are often built and run in experimental engines. If troubles occur, parts are redesigned and improved to a point where they will withstand operating loads. The lightest design would have each part sufficiently strong for its application with no excessive material. The procedure of durability testing works toward this end for the reason that the weaker elements are gradually redesigned and strengthened until all parts are compatible. This testing is ultimately carried to a point where the complete engine is capable of withstanding overloads far beyond operating conditions expected to be encountered in service. *From a paper by Earle A. Ryder, consulting engineer, Pratt and Whitney Aircraft Division, at the SAE Summer Meeting, June,* 1950, *in French Lick, Ind.*

XUM

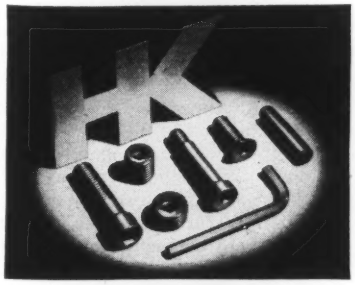
ANOTHER HOLO-KROME FIRST!

HK has the button head

Holo-Krome Button Head Socket Cap Screws are the newest addition to the famous H-K family of Socket Screw Products! Like all H-K products, they are designed and engineered to fill a definite need in Industry. Completely cold forged by Holo-Krome's exclusive, patented process, H-K Button Head Screws have *thin heads* which give clean lines, lower protuberance . . . a real advantage when countersinking is impractical in your product design. Sockets, with a true hex shape, well defined corners, smooth taperless walls, retain their original shape *without deformation* under continuous tightening and loosening . . . eliminating replacements and danger from sharp, burred heads. These advantages save money and *speed up your assembly operations.*

For better Button Heads . . . specify Holo-Krome!

All standard H-K FIBRO FORGED Socket Screw Products are manufactured from special analysis alloy steel scientifically heat treated to develop the utmost in necessary physical properties. Constant check of all materials and production line samples in Holo-Krome's own laboratories maintain the high standards which are your ***guarantee of unfailing performance.***



HOLO-KROME

Completely Cold Forged GUARANTEED UNFAILING PERFORMANCE *Socket Screw Products*

THE HOLO-KROME SCREW CORP., HARTFORD 10, CONN.

FRACTIONAL HORSE POWER

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INTERNAL,
HELICAL, SPUR
ELLIPTICAL, BEVEL
and WORM GEARS.
WORMS, RATCHETS
and SPROCKETS.

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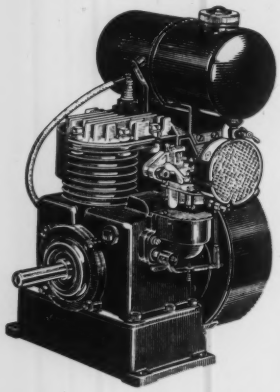

Model K7-1, *1½ H.P. 4-cycle, single cylinder, air-cooled. Length 13", width 14", height 17", Weight 38 lbs.*

POWER FOR:

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Garden Tractors
Water Pumps
Concrete Mixers
Construction Equipment
Conveyors
Rollers
Compressors
Sprayers
Motor Bikes
Scooters
And other uses.

Four-Cycle, Air-Cooled

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Compact • Quick-Starting Reliable

Kohler Engines are designed, engineered and built to give the same dependable, economical service that has won world-wide acceptance for Kohler Electric Plants in construction and other industries. Made in several sizes, they are suited to a wide range of uses. Kohler engineers will gladly assist in working out applications for your particular requirements. Write for information. Dept. 16-W, Kohler Co., Kohler, Wisconsin. Established 1873.

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PLUMBING FIXTURES • HEATING EQUIPMENT • ELECTRIC PLANTS • AIR COOLED ENGINES

NEW MACHINES

And the Companies Behind Them

Domestic

AUTOMATIC DISHWASHER: Front opening provides unbroken table top. Uses 825-watt heating coil for drying. Cycle includes 45-second spray, two 5-minute washes, two 1-minute rinses, and 30-minute drying period. Unit measures 40 in. high, 25 in. deep, 27 in. wide. Powered by 115-v ac motor. *Hotpoint Inc., Chicago, Ill.*

ELECTRIC WATER HEATER: Automatic unit has C-shaped heating element that can be connected for 1000, 2000 or 3000 watts. Elements located in lower part of tank, can be rotated to any of three positions. In upper position, tank heats 50 gal; in horizontal position tank heats 60 gal; in lower position tank heats 80 gal. *A. O. Smith Corp., Kankakee, Ill.*

ELECTRIC WATER HEATERS: Feature dial for selecting desired temperature. Serviceman no longer required to make adjustment. In 30, 40, 52, 66, and 82-gal cylindrical models; 30 and 40-gal table-top models that match kitchen appliances; and 52-gal square model. *Hotpoint Inc., Chicago, Ill.*

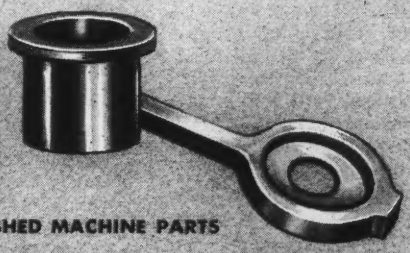
Heating and Ventilating

AIR CONDITIONER: Window fan cooler for homes, hotels, offices, etc. Filters, washes, and circulates outside air, dropping air temperatures between inlet and outlet to 20 degrees. Handles 2600 cfm air. Requires no water or drain connections, weighs 46 lb, measures 23½ in. high by 24 in. wide by 15 in. deep. With or without coin operating mechanism. *Kool-Sleep Inc., Cambridge, O.*

AIR CONDITIONERS: New line includes 2, 3, 5, 7½, and 10-hp capacity sizes. For offices, homes, stores, restaurants, etc. Three smaller sizes styled for in-the-room use. All units completely packaged. All except 2-hp size can be used for winter heating by addition of accessory steam or hot water coil. *General Electric Co., Bloomfield, N. J.*

EVAPORATIVE AIR CONDITIONER: Automatically adjusts evaporation rate with temperature change. Cooling area divided into five surfaces which are wetted one-by-one as dry bulb temperature increases. Water flow to mats controlled by bellows sensitive to dry bulb temperature;

QUALITY



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★ Batteries of presses and other equipment for quick service.
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★ 3000 additional engineers for consultation on technical problems.
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Cost conscious purchasing agents, engineers and production men examining their sources of supply will do well to consider Accurate as a source for springs, wire forms and small stampings. We at Accurate have an enviable record of saving money for our customers. Our production "know-how" is backed by a modern plant equipped with the very latest cost-cutting springmaking machines to produce uniformly accurate components for your product.

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Springs, Wire Forms, Stampings

second bellows, responsive to wet bulb temperature, shuts off water to mats if humidity reaches upper comfort level. *United States Air Conditioning Corp., Minneapolis, Minn.*

Maintenance

FLOOR MACHINES: For all floor preparation and maintenance, including scrubbing, waxing, polishing, sanding, steel wooling, pumicing, etc. Entire weight of machine (79 to 112 lb, depending on model) rests on brush. With brush revolving and bristles flush with floor, machine remains stationary. Tilting provides self-propelled movement. Features automatic brush coupler, rotary handle-grip switch, self-retracting wheels, solution-flow regulator. *Breuer Electric Mfg. Co., Chicago, Ill.*

BATTERY CHARGERS: For electric industrial trucks. For both lead-acid and nickel-iron-alkaline batteries. Include battery cutoff and charger shut down which operate when battery is fully charged, automatic disconnect if power is interrupted and automatic restart. Charging rate automatically controlled by modified constant voltage system. One model covers lead-acid units of 12 to 20 cells to 530 amp-hr capacity and nickel-iron batteries of 20 to 24 cells to 500 amp-hr capacity; other model for 12 to 20-cell lead batteries to 750 amp-hrs and nickel type units of 20 to 24-cells to 700 amp-hrs. *General Electric Co., Schenectady, N. Y.*

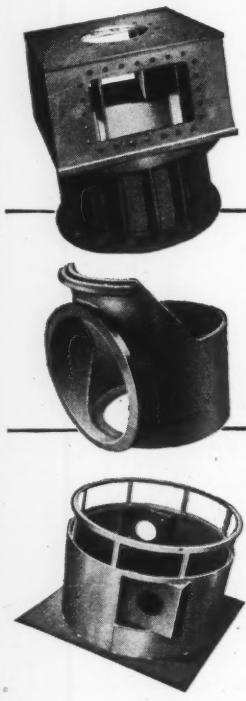
Manufacturing

BAND SAW: Uses spiral blade with 360-degree cutting edge which permits cutting in any direction. Spring tempered, hardened spiral blade has Rockwell hardness of C 81. Model 100A has 12-in. throat, 8½-in. clearance under guide bar, 4-in. blade tension adjustment, and table tilt of 45 degrees right, 5 degrees left. Powered by 1/3-hp motor. Model 500A powered by ½-hp motor, features variable-speed drive and planetary transmission to give speed range of 70 to 5000 fpm. *Tyler Mfg. Co., New York, N. Y.*

WET CUTOFF MACHINE: Swing-cut machine for metals. Equipped with recirculating pump, tank, hose connections and splash guards. Wheel depth control arm adjusts depth of cut. Adjustable work stop used for pieces to 24 in. long. *Stone Machinery Co. Inc., Manlius, N. Y.*

SHEAR: For sheet metal squaring and shearing. Knife cuts at constant, hydraulically-controlled speed. In 12

Weldments by Littleford



EXPERIENCE IS YOUR ASSURANCE OF QUALITY WORK AT A LOW COST

Design Engineers have found Littleford's experience in shearing, flame cutting, punching, bending, rolling and welding plate and sheet metals a great boon in developing new products. Fabricated Weldments by Littleford have found a place in almost every industry. Engineers know too that Littleford fabricates with unerring precision and assumes a definite responsibility for the quality of the finished product. If you have a Weldment problem and want this Littleford experience and "know how" to work for you, send blueprints for an estimate of cost or write for the Bulletin on Littleford Weldments.

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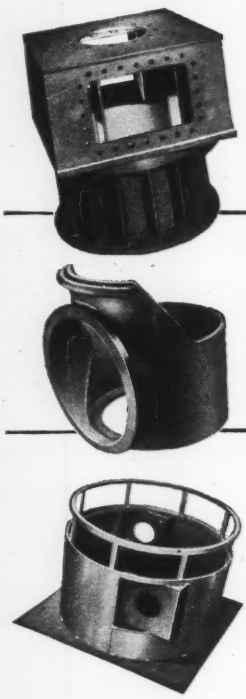

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PACKARD MOTORS

Your assurance of superior fractional horsepower performance

You can count on Packard fractional horsepower motors to provide top performance and dependable service. They will build customer good will . . . increase the value and trade acceptance of your products.

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DEPENDABLE APPLIANCE MOTORS FOR THIRTY-THREE YEARS

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for precision metal parts made by

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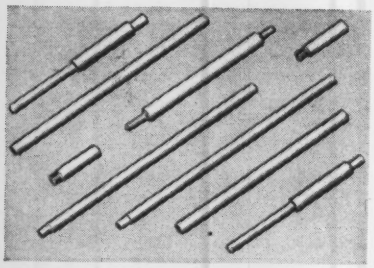

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TORRINGTON NEEDLE BEARINGS

sizes ranging from 10 gage to 5/16-in. shear. Ten foot, 10-gage model has only 7 moving parts, weighs 8600 lb, measures 131 in. long by 53 in. wide by 66 in. high. *Johnson Machine and Press Corp., Elkhart, Ind.*

SQUARING SHEARS: Power models for small sheet metal shop, heating and air conditioning contractor, etc. Cutting lengths of 3, 4, 5, and 6 feet available. Handles 6-ft lengths of 14 gage sheet at 85 cuts per minute. Cutting knives have four edges, can be rotated 90 degrees to a new edge, resulting in little down time. Can be furnished with electric foot switch. *Niagara Machine & Tool Works, Buffalo, N. Y.*

HYDRAULIC DRAW PRESS. For drawing or forming of panel corners, one corner at a time. Eliminates need for hand welding, metal finishing and separate dies for each size panel. Handles most radii and any panel size from 15 by 18 in. up. *Vulcan Tool Co., Dayton, O.*

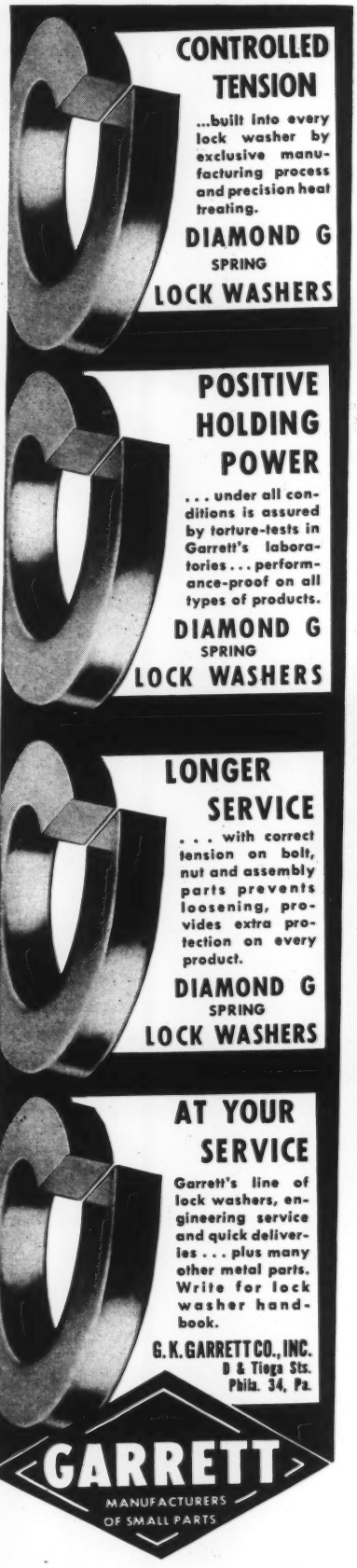
SOLDERING TOOL. New 1000-watt model operates on resistance heating principle. Touching work with tool completes secondary power circuit. Draws current only when in contact with work. *Ideal Industries Inc., Sycamore, Ill.*

GEAR HOBBING MACHINE: Automatic hob shifter now available on 16-16 model, shifts in increments down to 0.0026-in. with resulting increased tool life as high as 50 per cent. Hob slide constructed to eliminate lodging of chips between gibs and ways. Bed incorporates one flat and one V-way. Electrically-actuated jaw clutch engages feed with preset stops on feed control selector actuated by limit switches. Feed, index and differential gears all identical, with 30-degree involute splined holes. Speed change gears, also having 30-degree involute splined holes, are helical for quiet operation. *Barber-Colman Co., Rockford, Ill.*

Materials Handling

ELECTRIC FORK TRUCK: Capacity, 1000 lb. Features automatically accelerated control, 30 per cent higher speed, dead man brake operating off drive shaft which sets when seat is empty. Direction of travel cannot be changed until motor comes to full stop. Separate motor on hydraulic pumps results in 80 per cent increase in lifting and tilting speed. Lift heights from 84 to 130 in. with respective overall heights of 61 and 84 in. *Clark Equipment Co., Industrial Truck Div., Battle Creek, Mich.*

LOAD STABILIZER: For full-pallet handling of soft drink bottle cases. Con-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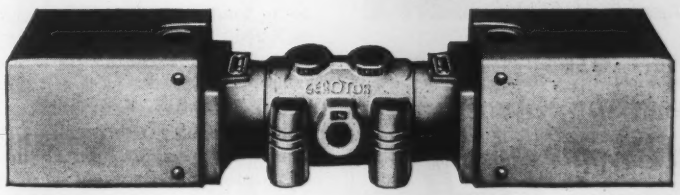
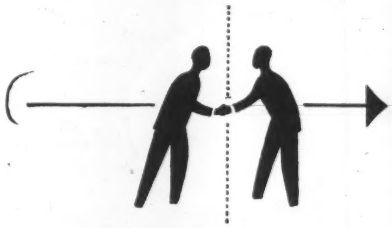


Rivett Series 5550 Double Solenoid Operated Spring Centered Valve.



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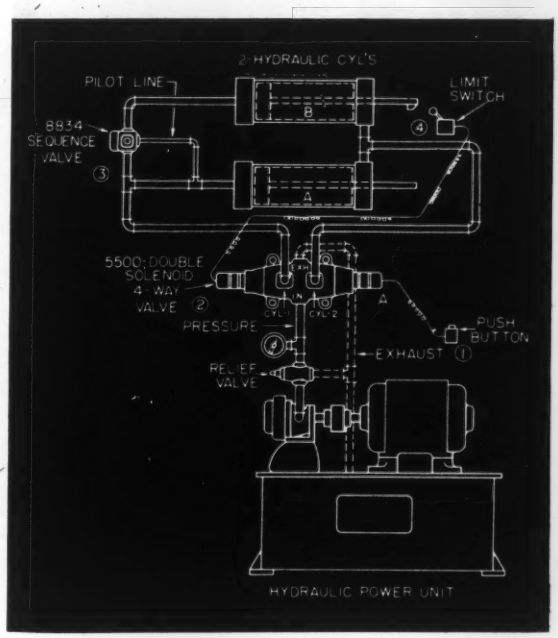
You can be sure of positive control of hydraulic power with the new Rivett Solenoid Hydraulic Valves.

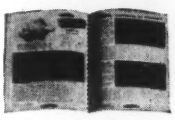
Designed with a hydraulically balanced piston, these valves are available in Double Solenoid with Standard Action, Single Solenoid with Spring Return, and Double Solenoid with Spring Centered. Solenoid Pilot Valves are also offered in each of the above three types of action. Five standard piston designs are offered to meet the requirements of any hydraulic circuit; other arrangements of spool porting may be had.

Features that make these valves outstanding in performance include balanced piston construction; a one piece, hardened, ground and lapped plunger; "O" ring seals; solenoid covers; symmetrical design.



SEMI-AUTOMATIC ELECTRIC CONTROL WITH SEQUENCE FEATURE

Cycle is started by momentarily depressing push button 1 which energizes solenoid coil of Model 5500 4-way control valve 2. Cylinder A extends; at completion of outward stroke, pressure will increase in cylinder A supply lines causing sufficient pilot pressure to open sequence valve 3, permitting oil to flow to cylinder B, and cylinder B extending. Limit switch 4 is momentarily actuated by piston rod of cylinder B, energizing opposite solenoid coil of 4-way valve. Oil flow to cylinders is reversed and cylinders A and B retract. Where piston velocities and inertia of connected loads are high, cushion cylinders are recommended. If external stops are provided, cushioning is not required.



Write for 50-page Catalog Section No. 202 illustrating and describing all Rivett Hydraulic Valves.

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WHEN YOU APPLY HYDRAULIC OR AIR POWER *Plan with*

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AIR AND HYDRAULIC VALVES AND CYLINDERS, HYDRAULIC POWER UNITS

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Ledeen Mfg. Co.
1606 S. San Pedro
Los Angeles 15, Calif.

sists of flat metal plate mounted on special carriage extension of standard lift truck. Double-acting hydraulic cylinder clamps load between plate and forks. *Towmotor Corp., Cleveland, O.*

ELECTRIC TIERING TRUCK: Line includes skid-tiering model, pallet-tiering models for single-faced pallets and for double-faced models. Pallet models handle 2000 lb at 24-in. load center; skid model handles 2000 lb or 4000 lb. Trucks tier 48 by 48-in. loads at right angles from 6-ft aisle. *Lyon-Raymond Corp., Greene, N. Y.*

HORIZONTAL BELT CONVEYOR: Utility, knock-down type units include drive motor and control equipment, tail end with 4-in. diameter crowned-face tail pulley and tracking idler, H type stands with return idlers, and choice of belt and 10-ft sections of roller or slider bed conveyors. Belts available in 8 to 36-in. widths in 2-in. increments. Standard conveyor speed, 50 fpm, with variable-speed drive available. *Sage Equipment Co., Buffalo, N. Y.*

ELECTRIC FORK TRUCK: Capacity, 4000 lb. Medium-weight truck (7600 lb) lifts to base height of 130 in. in 26 seconds, has max loaded travel speed of 5½ mph. Column hydraulically tilted 5 degrees forward in 2 seconds, 12 degrees backward in 4 seconds. Overall length with 36-in. fork, 116 in.; width, 41 in.; height with uprights telescoped, 83 in. Has 74-in. turning radius in intersecting aisles, 134-in. turning radius in right-angle aisles. *Elwell-Parker Electric Co., Cleveland, O.*

TRUCK: Gasoline-powered carrier for railroads, industrial plants, warehouses, foundries, etc. Capacity, 2000 lb; loading area, 21 sq ft. Has 4-cylinder, 20-hp, water-cooled engine, governor-controlled for speeds to 20 mph. Includes three speeds forward, one reverse. Has single front steering wheel, two rear drive wheels. *Lewis-Shepard Products Inc., Watertown, Mass.*

Plant Equipment

AIR COMPRESSORS: Line of gas-engine-driven units in 4, 6 and 8 power-cylinder sizes rated at 110, 165 and 220 hp. Have 8¼-in. bore, 9-in. stroke, with two power cylinders for each compressor frame. Compressors are of four-cycle, V-angle design with power connecting rods articulated with compressor rods. Units are balanced for smooth operation and light enough for semi-portable use. *Ingersoll-Rand Co., Phillipsburg, N. J.*

SAW BLADE WELDER: Permits welding and annealing of saw blades. Blade ends aligned by cam operated

Here's a Better Bearing!

American Crucible methods, experience, know-how and equipment result in highest quality and attractive savings to you. Bearings, bushings and wearing parts, machined or rough cast from

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THE Engineered BRONZE

carry a
MONEY-BACK GUARANTEE OF LONGER SERVICE AND LOWER MAINTENANCE COST

Castings to your patterns—any size, shape or section up to 3,000 lbs. Pattern making, designing and machining.

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Round, solid or tubular. Rough cast or fully machined. Cored stock, all sizes (by ⅛" steps) from ½" minimum core to 26" O. D. and 13" lengths or less. Six grades of hardness.

Let us quote on your requirements and make recommendations as to alloys. Simply send blueprints, conditions of operation and other data.

THE AMERICAN CRUCIBLE PRODUCTS CO.
1321 OBERLIN AVE.
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FARVAL—*Studies in Centralized Lubrication* No. 117

Lubrication of sugar mill is automatic, adequate, safe with Farval

ANY sugar mill, new or old, benefits from Farval Centralized Lubrication. For Farval saves time and effort, increases machine efficiency, reduces bearing expense, and eliminates waste of lubricant.

At Sanalona, Mexico, the new Farval-protected mill illustrated here will not develop bearing trouble due to faulty lubrication. There will be no need to drench overheated bearings with buckets of oil, no delays or shutdowns to repair or replace bearings. The mill can run steadily throughout the grinding season.

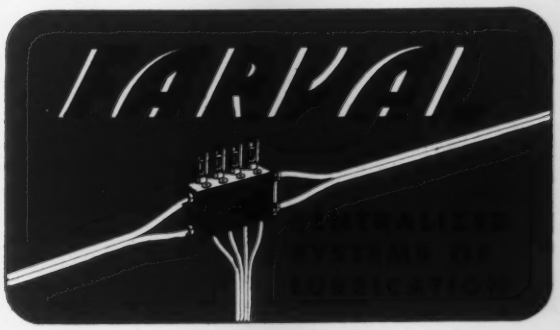
Older mills, too, can use Farval with profit. It is simple and easy to install a centralized system on any mill while it is down, during the off season check-up. And Farval pays for itself the first grinding season by the economies effected.

Farval is a practical system proved in 20 years of service. It is in widespread use throughout industry—on scores of sugar mills and on thousands of hot and cold rolling mills, presses, conveyors, cranes and other equipment in steel and metal working plants, chemical processing plants, food plants—in short, wherever bearings operate under heavy loads and need sure, adequate protection.

Farval is the original Dualine system of centralized lubrication. The Farval valve has only two moving parts—is simple, sure and fool-proof, without springs, ball-checks or pinhole ports to cause trouble. Through its full hydraulic operation, Farval unfailingly delivers grease or oil to each bearing—as much as you want, exactly measured—as often as desired. Indicators at every bearing show that each valve has functioned.

Write for Bulletin 25 for full details. The Farval Corporation, 3265 East 80th St., Cleveland 4, Ohio.

Affiliate of The Cleveland Worm & Gear Company, Industrial Worm Gearing. In Canada: Peacock Brothers Limited.



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Sliding sleeve for machine tool clutch. Made from sintered iron powder, oil-impregnated. No machining or finishing. More than ample strength.

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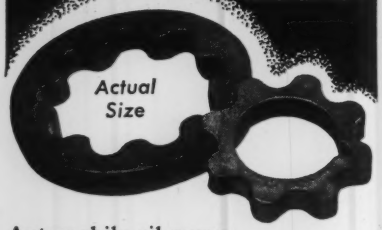

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jaws, weld selector provides correct spring tension to welding jaw for blade widths from 1/16 to 1/2-in. Pushbutton controls heat for annealing operation. Thickness gage below holding fixture tests for proper blade thickness at weld. Blade squaring shear and etching pencil available as accessories. *The DoAll Co., DesPlaines, Ill.*

WATER COOLER: Explosionproof electric cooler safe for operation in hazardous locations. Equipped with hermetically sealed refrigeration system, with stainless cold water storage tank, and seamless copper precooler. Dimensions, 14 by 15 by 43 in. Supplied with foot pedal or hand control to operate bubbler. *Cordley & Hayes, New York, N. Y.*

Power Plant Equipment

DIESEL ENGINES: New type engines in 195 to 375 hp range. Units are 4-cycle, 7-in. bore, 8½-in. stroke, 900 to 1000-rpm, single-acting models having weight of 30 lb/hp and specific fuel consumption of 0.40 lb/hp/hr. Cylinders have wet type liners, individual heads with overhead valves and intake and exhaust valve inserts. Main and crankpin bearings have aluminum alloy, full floating, interchangeable shells. Normal starting is by 250 psi air admitted to all cylinders in turn through starting-air distributer. Available in 6 or 8 in-line designs. *Ingersoll-Rand Co., Phillipsburg, Pa.*

SPECIAL-PURPOSE GENERATOR: Can be arranged to give as many as six different frequencies for experimental or production testing or where odd or multiple frequencies are required. Motor operates on three-phase ac of standard voltages and frequencies, or on 24 or 48-v dc. Originally designed as harmonic ringing assemblies for telephone service. *Kato Engineering Co., Mankato, Minn.*

Testing and Inspection

OPTICAL COMPARATOR: Features motor-driven unit for raising and lowering work table. Single switch controls vertical rapid traverse. Delicate final adjustment made by knob on motor shaft. *Jones & Lamson Machine Co., Springfield, Vt.*

GEAR CHECKER: Rolling fixture to indicate errors in size and eccentricity and to check for excessive roughness. Master gear held in slide carriage which is retracted by hand lever. Inspection made by advancing lever to mesh gears, errors being registered on dial indicator. *National Broach & Machine Co., Detroit, Mich.*

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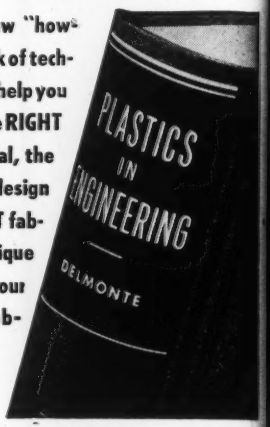

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